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by

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# TECHNICAL REPORT

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ABSTRACT

This report describes the design and development of a Gas Supply System for Hermetically Sealed Platforms. Immediate interest in this system is for future applications of the ST-124 stabilized platform for the Saturn V Instrument Unit.

The Gas Supply System is composed of (1) a Recirculation Subsystem which continuously compresses nitrogen from the platform and recirculates it to the gas bearings used in the AMAB-3 accelerometers and the AB-5 gyros, and (2) a Gas Make-Up Subsystem for feeding nitrogen into the Recirculation Subsystem at a rate sufficient to match any reasonable leakage rate from the circulating loop to space ambient.

The major accomplishment of the program has been development of a unique type of compressor for the Recirculation Subsystem. The compressor, referred to as a Resonant Piston Compressor, has been designed for (1) a minimum of 10,000 hours of continuous, maintenance-free operation, (2) zero contamination of the nitrogen gas, (3) hermetically sealable construction, (4) operation during launch, and (5) operation over an ambient temperature range of 0 to 60°C.

Preliminary testing of two compressors at MTI has successfully demonstrated the compressor concept. Additional long-term and environmental testing of the units is presently underway at NASA-MSFC, with more than 2700 hours of trouble-free operation now accumulated on one machine. One of the most significant aspects of compressor performance is the almost 60 percent overall efficiency (adiabatic) demonstrated at design-point flow and pressure rise. A maximum efficiency of almost 75 percent has also been demonstrated at a somewhat higher pressure rise. With better understanding of the operating principles of the compressor, and further reduction of internal leakages, it appears that efficiencies in the range of 80 to 85 percent should be attainable.

## I. INTRODUCTION

Mechanical Technology Incorporated, under NASA contract NAS 8-11660, has developed a "Prototype Gas Supply System for Hermetically Sealed Platforms". This work was performed for the Inertial Sensors and Stabilizers Division of the Astrionics Laboratory, George C. Marshall Space Flight Center.

The Gas Supply System constitutes a means for providing a 10,000 hour supply of pressurized nitrogen to the gas bearings used in the AMAB-3 accelerometers and the AB-5 gyros for the ST-124 stabilized platform. To accomplish this function, the Gas Supply System is composed of two major subsystems as follows:

1. a Recirculation Subsystem which continuously recompresses the nitrogen from the discharge side of the bearings and recirculates it to the inlet side of the bearings;

2. a Gas Make-Up Subsystem comprised of a small high-pressure nitrogen reservoir and a means for feeding gas from the reservoir into the Recirculation Subsystem at a rate sufficient to match any reasonably small out-leakage rate from the circulating loop to space ambient.

The heart of the Recirculation Subsystem, and, in fact, of the whole Gas Supply System, is the nitrogen compressor. The requirements for this compressor are:

1. long, maintenance-free life (in excess of 10,000 hours continuous operation)

2. hermetically sealable construction

3. truly non-contaminating operation (less than one part per million by weight of condensable hydrocarbon content)

4. ability to operate during launch

5. ability to operate over an ambient temperature range of 0 to 60 C

6. minimum possible drive power

7. minimum weight and size consistent with the above objectives.

The above requirements are beyond the capabilities of any conventional or commercially available compressor. They may, however, be attainable using a non-conventional compressor if the compressor concept is based upon inherent satisfaction of items 1 and 3 above. This was the approach taken by MTI in the development of the compressor for the Gas Supply System.

## Description of MTI Compressor Concept

The MTI compressor concept is presently referred to as a Resonant Piston Compressor. Aerodynamically, the unit is a high-speed reciprocating compressor in which gas flow is controlled by an inlet port and a discharge valve. Mechanically, the compressor differs significantly from the usual crankshaft/connecting rod, or cam actuated, reciprocating designs. In essence, the piston is part of a mass-flexure system which is driven electromagnetically in a resonant mode of vibration. Crankshaft and connecting-rod, or cam and cam-follower, mechanisms are completely eliminated. The advantages which accrue from such a drive mechanism are as follows:

1. All relative motion bearings, both rotary and linear varieties, are completely eliminated, thus eliminating the need for lubrication of any kind in the piston actuation system.

2. Elimination of all relative motion bearings in the actuation system greatly reduces, of course, the total number of component drive parts. Inherent reliability is greatly increased provided peak stresses are well below the endurance limit of the flexure material.

3. Elimination of all relative motion bearings in the actuation system greatly increases the mechanical efficiency of the compressor, since the only mechanism losses are now internal material damping in the flexures and external windage effects, both of which are very small in a resonant flexural system.

While it has been possible to eliminate bearings from the piston actuation system, there must still be a linear bearing between the piston and the cylinder in order to maintain a seal around the piston as the gas is compressed. The usual approach to this sealing problem has been to use piston rings which can be classified as follows:

1. oil lubricated rings, which are unacceptable in this application for reasons of contamination,

2. unlubricated teflon rings, which again are unacceptable because of limited life capability, and

3. unlubricated carbon rings which, while being relatively successful, generally require a conventional piston actuation system using grease-packed ball bearing elements.

The MTI approach to piston sealing is to provide a very close clearance between the piston and cylinder wall. The relative motion of the piston with respect to the cylinder wall is then used to generate a hydrodynamic gas-bearing action to center the piston, or alternately, a small bleed flow can be taken from the compressor discharge to pressurize the space between the piston and cylinder walls thus providing a hydrostatic gas-bearing centering action. In either case, the actual gas being compressed, i.e., the system fluid, is the fluid which is used to lubricate the piston. The close clearance between piston and cylinder reduces compressor leakage to small values, while the gas-bearing action eliminates rubbing contact between the piston and cylinder walls. As a result, a very low-friction, long-lived, contamination-free piston sealing means is obtained.

Status of Gas Supply System Development

Contract NAS 8-11660 required delivery of one complete Prototype Gas Supply System plus one spare compressor. The first compressor for the Recirculation Subsystem was shipped to MSFC in February, 1966. The spare compressor and the Make-Up Subsystem were shipped in June, 1966.

No difficulty was encountered in meeting the compressor flow versus pressure rise requirement given in the "Scope of Work" for NAS 8-11660. Likewise, there have been no mechanical failures in either of the two compressors to date. Both compressors were operated for more than 110 hours at MTI before shipment to MSFC. The significance of these test hours lies in the fact that the "knee" of the S-N curve for both the discharge valve and the resonant spring material is reached in about $1 \times 10^6$ cycles of piston displacement, which corresponds to 4.63 hours of operation. Thus the 150-plus test hours accumulated at MTI was more than sufficient to demonstrate successful operation of the basic compressor concept. Beyond this point it is a question of long-term effects, which can only be identified by long-term operation. Such long-term operation is presently being accumulated by MSFC. As this report is being written, the first and second delivery compressors have accumulated over 2700 and 500 hours respectively of trouble-free operation at MSFC.

In addition to failure-free operation to date, compressor design flow and pressure rise have been obtained with approximately 72 watts electrical input power to the drive solenoid. This gives promise that the total power drain on the vehicle power supply (which will include compressor drive circuit losses) will be much less than the 150 watt maximum allowed in the "Scope of Work" for NAS 8-11660.

The electrical drive concept for the compressor utilizes solid-state switching circuitry to generate a 30 hz sinusoidal current waveform in the solenoid coil from a 28 VDC power supply. Although MTI did not demonstrate operation of the drive circuit at full compressor output, it is understood that MSFC has obtained full-power performance of the concept by a redesign of the transformer which couples the switching circuit to the drive solenoid. The compressor can also be driven directly from a 60 hz a-c power supply by simply inserting a diode in series with the solenoid coil.

The following sections of this report document the design of the Prototype Gas Supply System. While environmental testing of the compressor at MSFC has identified some deficiencies in the compressor mounting system, MTI believes that these deficiencies can be corrected by simple modification to the mount design. The modifications are discussed in Section III of this report. MTI has also recommended a slight rearrangement of the platform $\Delta P$ regulating components to obtain a better match between compressor operating characteristics and $\Delta P$ regulation requirements. This rearrangement, which has been confirmed by MSFC tests, is described in Section V.

## II. SYSTEM DESIGN

### System Specification

The original specification for the Gas Supply System is given in Exhibit A of contract NAS 8-11660. The first several pages of Exhibit A contain the technical aspects of the specification and are included in Appendix A of this report for reference purposes.

### System Configuration

Subsequent to careful review of the System Specification and preliminary discussions with several component vendors, MTI recommended a system configuration as shown in Figure 1. This configuration differs from the reference configuration of Exhibit A (shown on page A-3) in the following aspects.

1. An absolute pressure regulator (item S of Figure 1) is used in the Gas Make-Up Subsystem in place of the pressure-switch actuated solenoid valve. The advantages which accrue are

   a) the platform ambient pressure (which is also compressor inlet pressure) can be held within $\pm$ 0.5 psi, as compared to $\pm$ 1.5 psi for the pressure-switch/solenoid-valve combination;

   b) the single pressure regulator should be more reliable than the pressure-switch/solenoid-valve combination.

2. A relief valve (item R of Figure 1) is provided to maintain platform ambient pressure between 17 and 19 psia in the event that the absolute pressure regulator (item S) should fail in an open position.

3. A calibrated orifice (item J of Figure 1) is used to limit flow from the high-pressure nitrogen reservoir in the event that the absolute pressure regulator (item S) should fail in an open position.

The Recirculation Subsystem as shown in Figure 1 is basically the same as shown in the reference configuration of Exhibit A (shown on page A-3) except that (1) an additional gas cooler is used to cool inlet gas to the compressor, (2) the liquid coolant for the two gas coolers also flows around the compressor drive solenoid to remove $i^2R$ coil losses, and (3) a restrictor (item F of Figure 1) is placed in the low-pressure side of the differential pressure regulator (item E) to limit by-pass leakage flow around the platform in the event that the regulator diaphragm should fail.

## Component Specifications

Subsequent to approval of the Gas Supply System concept as shown in Figure 1, a specification was prepared for each of the system components which would be purchased from outside vendors. These component specifications are included in Appendix B of this report for reference purposes.

## III. COMPRESSOR DESIGN

The compressor for the Gas Supply System is a single stage, reciprocating, resonant piston compressor with a valved discharge and a ported inlet. The valved discharge improves the efficiency of the compressor and makes the design more predictable from both an aerodynamic and mechanical standpoint. The ported inlet, while being less efficient than a valved inlet, provides a very convenient means of getting the required inlet flow area with the utmost of simplicity and reliability. The unique features of the compressor — that is, the resonant vibration mode of operation and the elimination of all bearings except for the close-clearance gas-lubricated piston seal bearing — are described in the introductory section of this report.

### Aerodynamic Design

MTI computer program PNO 195 was used to converge to a set of compressor dimensions which would produce the required compressor performance over an inlet pressure range of 12 to 15 psia. Approximately 30 compressor designs were reviewed during the convergence process. The design constraints were:

a. platform pressure-flow requirements,
b. minimum physical size,
c. minimum cycle power.

Compressor flow requirements were determined from MSFC curves for the AMAB-3 accelerometer and the AB-5 gyro, dated September 3, 1964. This flow data is replotted in Figures 2 and 3 as a function of bearing $\Delta P$ for the two limits of platform ambient pressure, $p_a$. The component flow requirements as determined from Figures 2 and 3 are summarized in Table I below.

TABLE I

REPRESENTATIVE FLOW REQUIREMENTS FOR ONE AMAB-3 ACCELEROMETER AND ONE AB-5 GYRO

| Ambient Pressure psia | Flow - std. cc./min. | | | |
|---|---|---|---|---|
| | ΔP (min.) = 14.5 psi | | ΔP (max.) = 15.5 psi | |
| | AMAB-3 | AB-5 | AMAB-3 | AB-5 |
| 12.0 | 1960 | 1590 | 2115 | 1725 |
| 15.0 | 2215 | 1795 | 2395 | 1945 |

Since there are three accelerometers and three gyros in each platform assembly, the total platform flow (neglecting leakage) becomes as shown in Table II.

TABLE II

RANGE OF TOTAL REQUIRED PLATFORM FLOW (NEGLECTING LEAKAGE)

| Ambient Platform Pressure - psia | Flow - SCFM | |
|---|---|---|
| | ΔP (min.) = 14.5 psi | ΔP (max.) = 15.5 psi |
| 12.0 | 0.376 | 0.407 |
| 15.0 | 0.424 | 0.460 |

To arrive at the compressor bore/stroke dimensions, estimates of platform and compressor leakages were added to the above flow requirements, and an estimate of the additional recirculating-loop pressure drop was added to the required platform ΔP. Reiterative use was then made of computer program PNO 195 to converge to a set of compressor dimensions which would best satisfy the flow-pressure requirements over the range of 12 to 15 psia inlet pressure. A summary of the calculated performance for the selected compressor design is given in Table III. Predicted compressor p-V diagrams for the 12 and 15 psia inlet pressure conditions are shown in Figures 4 and 5 respectively. These diagrams are not exact in that the p-V characteristic during the inlet interval has been approximated rather than calculated. However, the approximation is conservative from a total cycle power standpoint.

It is seen from Table III that there is considerable flow margin in the compressor design to allow for internal platform and compressor leakage, and reasonable inaccuracies which may exist in the compressor design (such as estimation of discharge coefficients, etc.). In terms of pressure rise, Table III shows that appreciable pressure margin is obtained as inlet pressure approaches 15 psia. As inlet pressure approaches 12 psia, however, the predicted pressure margin approaches zero, i.e., the compressor pressure rise will just equal the expected line and regulator drops, plus the 15 psi platform drop. The low inlet pressure condition is therefore the more critical operating condition, with pressure rise being the prime concern.

TABLE III

CALCULATED COMPRESSOR PERFORMANCE

Piston stroke - 0.5 in.

Piston diameter - 1.69 in.

Compressor speed - 3,600 spm

Compressor inlet temperature - 70 F

Gas - nitrogen

| Compressor inlet pressure - PSIA | Required platform flow at 15 psi $\Delta P$ - SCFM | Compressor design flow - SCFM | Flow margin for leakage, etc. - PERCENT | Cycle pressure rise at design flow - PSI | Estimated total system pressure drop (includes compressor valve, platform, line, regulators, etc.) - PSI | Pressure margin PERCENT | Cycle power (does not include drive circuit losses) - WATTS |
|---|---|---|---|---|---|---|---|
| 12.0 | 0.392 | 0.47 | 19.9 | 19.0 | 19.0 | 0.0 | 40.6 |
| 15.0 | 0.443 | 0.59 | 33.2 | 23.0 | 19.0 | 21.0 | 50.0 |

Discharge Valve Design

The discharge valve consists of four cantilever reeds which are elastically deflected to an open position when cylinder pressure is higher than discharge manifold pressure. The reeds were sized by a conventional design procedure whereby valve $\Delta P$ is calculated by matching the time rate of change of mass in the cylinder to mass flow rate through the valve. This is an interative and incremental type of calculation which is further complicated by the fact that valve flow area is also a function of valve $\Delta P$.

The goal of the valve design effort was to limit valve $\Delta P$ to less than 3 psi. Several trys were required to find a suitable combination of valve dimensions which would satisfy all the design criteria. Since the $\Delta P$ calculations were based on only a three increment division of the discharge interval, and since only a limited number of manual iterations were carried out at each calculation point, a fáir amount of conservatism was used throughout the design effort to assure that valve $\Delta P$ would indeed be less than 3 psi.

Recently, in connection with another development effort, a digital computer program has been written for calculation of valve performance. This program has been used to check the original design calculations for the subject valve. The pertinent performance characteristics, based on the computer solution, are given in Table IV below. Notice that maximum valve $\Delta P$ is indeed considerably less than 3 psi which confirms the conservatism of the original manual solution procedure.

TABLE IV

DESIGN AND PERFORMANCE CHARACTERISTICS OF DISCHARGE VALVE

Number of reeds - 4

Thickness of reeds - 0.0098 inches

Reed natural frequency - 394.4 cps

Maximum reed stress, open position - 21,500 psi

Maximum reed stress, closed position - 3,338 psi

$\Delta P$ to fully open valve - 0.8 psi

Maximum valve $\Delta P$ during discharge - 1.7 psi

Maximum valve flow area (4 reeds) - 0.103 $in^2$

Reed material - 17-7 PH stainless, heat treated to CH900 condition

A picture of the valve components is shown in Figure 6.

Resonant Spring Design

The mechanical spring assembly which tunes the piston is the most critical component of the compresssor from the standpoint of life. Consequently, every effort was made to assure that the spring would not fail.

The spring assembly consists of four U-shaped springs. The U-springs are oriented 90 degrees apart with the ends of each spring being electron-beam welded together to form a one-piece assembly as shown in Figure 7. One end of the spring assembly is fastened to the "stationary" compressor cylinder; the other end is fastened to the reciprocating piston assembly.

In the original concept of the resonant compressor, the ends of the U-springs were shown as being mechanically clamped between pressure plates, rather than welded together. This type of clamp, however, always raises two questions:

1. What is the stress concentration factor which should be applied to the nominal spring stress at the clamping line?

2. How can fretting corrosion between the spring and clamp faces be reduced to a safe condition?

Both of these questions are difficult to answer, and in general can only be determined by comprehensive testing. It was decided, therefore, to eliminate these uncertainties by eliminating the mechanical clamp concept. Each of the U-springs was instead designed to have thickened end-plates formed by machining the center section of a single piece of flat spring stock down to the required thickness for the basic spring section. A generous radius was used to make the transition from the effective spring thickness dimension to the end-plate dimension. The use of integrally machined end-plates and a generous transition radius reduces the stress concentration factor at the spring clamping line to effectively 1.0. The problem of fretting corrosion is likewise eliminated since there are no highly-stressed clamped surfaces. The weld joints are made in the thickened end-plates and hence are subjected to very low stresses.

The spring material is 17-7 PH stainless steel heat treated to give a fatigue strength of 90,000 to 100,000 psi. Figure 8 shows the design map which was used to determine final dimensions of the springs. Actually, three such maps were made, one each for spring radii of 2.5, 2.625 and 2.75 inches. The 2.625 inch radius was selected on the basis of mechanical layout considerations. Figure 8 shows the combinations of total spring width (nb), spring thickness (h), spring length to radius ratio (L/R), and resulting maximum spring bending stress ($\sigma$), required to resonate a 0.87 pound reciprocating mass at 60 hz with a 0.25 inch amplitude. In order to maintain a fatigue stress safety factor of at least two, maximum allowable spring stress was designated as 40,000 psi. The selected values of nb, h, and L/R were 4.5 inches, 0.094 inches, and 0.665 respectively. The final design of the spring, as shown on MTI drawing 141D01, should give essentially infinite life at the design stroke conditions.

## Piston Bearing Design

In order to maximize compressor efficiency, self-acting gas-lubricated piston bearings were selected. Since the design flow of the compressor is quite small, the use of conventional hydrostatic gas bearings, pressurized from compressor discharge, would have resulted in about a 45 percent reduction in net compressor efficiency.

The configuration of the self-acting piston bearings is shown in Figure 9. It is seen that each end of the piston sleeve is machined with a very fine taper on the OD. These tapers, together with a short straight section of piston OD, constitute the upper and lower piston bearings as identified in Figure 9. The upper bearing has, for all practical purposes, compressor inlet pressure as ambient pressure at each of its ends. Furthermore, the length of the straight section of the bearing varies with displacement of the piston. The lower bearing always has cylinder pressure as the boundary condition at the end of its straight section and compressor inlet as ambient pressure at the entrance to its tapered section. The length of the lower bearing is constant.

The radial and angular stiffnesses of the combined upper and lower bearings have been analyzed as a function of ambient pressure ($p_a$), radial clearance (C), and amount of taper ($\beta C$) for the concentric position of the piston in the cylinder (nomenclature is defined in Figure 9). The analysis takes into account the variable length of the upper bearing, and the variation of cylinder pressure as given in Figures 4 and 5. Preliminary studies of the effect of parameter $\beta$ indicated that maximum values of radial and angular stiffness would be obtained with $\beta$ approximately equal to 1.0. Consequently, the final calculations were made for a range of $\beta$ values from 1.0 to 1.5 .

Figure 10 shows a plot of radial stiffness which is applicable for ambient pressures from 12 to 15 psia. Figure 11 shows angular stiffness for an ambient pressure of 15 psia. For 12 psia ambient, the stiffness values are approximately 25 percent lower. The ordinates of these stiffness plots run from zero to $2\pi$ radians which represents one complete cycle of piston motion. Zero and $2\pi$ radians represents maximum expansion or "bottom dead center" position, $\pi$ is maximum compression or "top dead center", while $\pi/2$ and $3\pi/2$ are mid-stroke positions.

It is seen from Figure 10 that the radial stiffness is positive during the compression stroke and negative for the expansion stroke. Furthermore, the stiffness function is not affected by ambient pressure or amount of taper over the range investigated. Stiffness is a strong function of clearance, however, with higher values obtainable at the smaller clearances.

Angular stiffness is seen from Figure 11 to be positive from zero to about $3\pi/2$ radians and slightly negative for the last quarter of the stroke. Angular stiffness is significantly affected by all the variables over the range investigated, that is, by ambient pressure, clearance, and amount of taper.

The existence of negative stiffness values during portions of the cycle immediately raises questions about the stability and load carrying capability of the piston bearing. These questions were not theoretically persued, nor was any attempt made to optimize the bearing design. It was noted that both radial and angular stiffness are positive during the compression stroke, when differential pressure forces acting on the piston are highest. During the last half of the expansion stroke (from $3\pi/2$ to $2\pi$), when total radial and angular stiffnesses are negative, there is in fact positive radial stiffness at the upper bearing. This, together with the considerations that (1) differential pressure forces are low during the last quarter cycle, (2) some finite time would be needed to accelerate the piston under the influence of the negative force gradients, and (3) piston velocity approaches zero as stroke approaches $2\pi$, suggested that if contact should occur, it most likely would occur (1) at the lower bearing, (2) at light loads, and (3) at low velocities. By selecting materials for the piston and cylinder having a proven high compatability and low wear rate under rubbing conditions, it was felt that a successful bearing might be obtained. Furthermore, there seemed to be a reasonable chance that contacts would not occur. With these considerations in mind, it was decided to proceed with a bearing design having a 0.5 to 0.8 mil radial clearance and a $\beta$ value of 1.0 to 1.5.

## Drive Solenoid

The drive solenoid is a simple electromagnetic device which can develop an axial attractive force, $\mathcal{F}$, according to the equation:

$$\mathcal{F} = \frac{k\Phi^2}{A} = k\beta^2 A$$

where $\Phi$ is total axial flux passing through the pole face area, A, of the plunger, $\beta$ is flux density, and k is a suitable constant depending on the system of units. The basic configuration of the solenoid is shown in Figure 12.

The required amplitude of the compressor-frequency harmonic driving force can be closely calculated from the equation

$$F = \frac{17.7P}{\omega a}$$

where

$F$ = harmonic force amplitude - lb.

$P$ = total average compression power - watts

$\omega$ = compressor frequency - rad./sec.

$a$ = piston displacement amplitude - inches.

For the present design, total compression power at 15 psia inlet pressure was calculated to be 50 watts. For a 0.25 inch piston amplitude and a 60 hz (377 rad./sec.) compressor frequency, the resulting 60 hz harmonic force amplitude was 9.38 lb.

Since the simple solenoid of Figure 12 always produces an attractive force, regardless of the direction of flux (i.e., of the direction of current), a 60 hz harmonic force component will always be accompanied by a d-c force component. The magnitude of the d-c component will depend upon the manner in which the 60 hz component is obtained. The simplest technique is to insert a diode in series with the coil and excite the circuit with a 60 hz voltage. Assuming that the coil is highly inductive, such that the time constant of the coil (L/R) is greater than 0.1 seconds, the resulting current waveform will be

$$i \cong I + I \sin(2\pi 60t).$$

Assuming a constant solenoid air gap (equal to the average air gap during one cycle of piston motion), the solenoid must be sized to produce a peak force $\mathcal{F}$ equal to 2F. The d-c solenoid force will be equal to F.

A second means to obtain the 60 hz driving force is to excite the solenoid with a 30 hz alternating voltage. In this case the current waveform will be simply

$$i = I \sin(2\pi 60t).$$

However, the solenoid must be sized to produce a peak force $\mathcal{F}$ equal to 2.355F, and the d-c force component becomes equal to 1.5F.

The 30 hz driving method was selected for the present application in order to permit improvement of the solenoid power factor (as seen by the drive circuit) by adding capacitance in parallel with the coil. Accordingly, the solenoid was sized to produce

$$\mathcal{F} = 2.355F = 22.1 \text{ pounds}$$

at an average air gap of 0.3 inches. The resulting measured force versus current characteristic for the solenoid design is shown in Figure 13. It is seen from Figure 13 that for a 0.3 inch gap, the solenoid just starts to enter its saturation region at about 22 pounds of plunger force. Figure 14 shows measured 60 hz rms volts versus amps for different plunger air gaps. Figure 15 shows a-c inductance of the solenoid, again as a function of plunger air gap.

## Drive Circuit Design

The concept for the compressor drive circuit is shown in Figure 16. It was desired to use a 28 VDC battery supply for the source of compressor drive power. Consequently, a D-C to A-C inverter was required.

The arrangement of Figure 16 uses a 30 hz square-wave signal generator to supply the reference frequency for the electrical circuit. A commercially available oscillator made by Connor-Winfield and having a $\pm$0.1 percent frequency stability was used. This signal generator (Model L118A) requires a 12 VDC input and has a $\pm$ 10 volt output into a 10K ohm resistive load. The output signal supplies the base current for the first stage of a three-stage transistor switching circuit. The third-stage power transistor switches the 28 VDC power supply across the primary side of a voltage step-up coupling transformer. The stepped-up voltage from the secondary of the coupling transformer is then applied to a parallel connection of the drive solenoid and a capacitor. The capacitor is selected to tune the solenoid drive circuit to 30 hz. The resulting current waveform through the solenoid is 30 hz sinusoidal. Tuning of the drive circuit also achieves a unity power-factor condition for the switching circuit and hence minimizes current levels in the power transistor and coupling transformer.

The above concept of the drive circuit was breadboarded and tested, and proven to be a workable concept. However, the secret of success for this circuit is the design of the coupling transformer. Two transformer designs were tested with only partial success in that only 75 percent of the desired compressor stroke was achieved. This was due to a lack of sufficient voltage from the coupling transformer.

Because of funding limitations, further work on the drive circuit was deferred in favor of completing the compressor and Gas Make-Up Subsystem development. Subsequent to delivery of the first compressor to MSFC, further work on the drive circuit was performed by the electronics group at the Astrionics Laboratory and successful, full power performance of the compressor was demonstrated with the 30 hz electrical drive circuit concept.

## Compressor Mounting System

The resonant compressor produces an alternating force whose peak amplitude is approximately 80 pounds. This force, which arises from alternate extension and compression of the resonant spring component, occurs at a fundamental frequency of 60 hz. It is considered imperative that this vibratory force be well isolated from the main IU mounting panel. (A quantitative definition of "well isolated", in terms of what constitutes an acceptable level of transmitted force, has yet to be determined).

To obtain the necessary force isolation, a low-frequency mounting system must be interposed between the IU panel and the compressor casing. This mounting system must be capable of effective attenuation of the vibratory force over a zero to 10-g range of constant acceleration in a direction parallel to the vibratory force vector. It is this latter condition of possible 10-g acceleration (during the Saturn V launch period) which is troublesome. The problem here is that good force isolation requires a very "soft" mounting system (i.e., a very low-frequency system) However, if the mount is too soft, impractically large "static" deflections of the isolation system will occur at the 10-g acceleration level. Hence, a trade-off must be made between magnitude of transmitted force and "static" deflection at 10-g.

The problem of mounting the resonant compressor thus consists of arriving at an optimum combination of mounting system stiffness and damping parameters such that

(1) the compressor vibratory force transmitted to the IU panel is attenuated to an acceptable level,

(2) excessively large static deflections are not encountered at the 10-g acceleration level,

(3) compressor performance is not significantly impaired as a result of the mounting system design, and

(4) launch vibration and shock transmitted from the IU panel to the compressor is reduced to levels which will not damage the compressor or cause more than a momentary impairment of performance.

The mounting system supplied with the compressors can be described as follows.

1. The primary, or soft, mount consists of three sets of coil springs equally spaced around the compressor. These springs completely isolate the compressor from the IU panel. The mount has a 3.7 hz vertical natural frequency and a somewhat lower lateral natural frequency (the vertical direction being parallel to the longitudinal axis of the launch vehicle). The primary mount should quite effectively isolate the compressor from vehicle launch vibrations. However, its main function is to attenuate the 60 hz, 80-pound force produced by the resonant spring. The dynamic force which will be transmitted through the soft mount to the IU panel is calculated to be $\pm 0.31$ pounds.

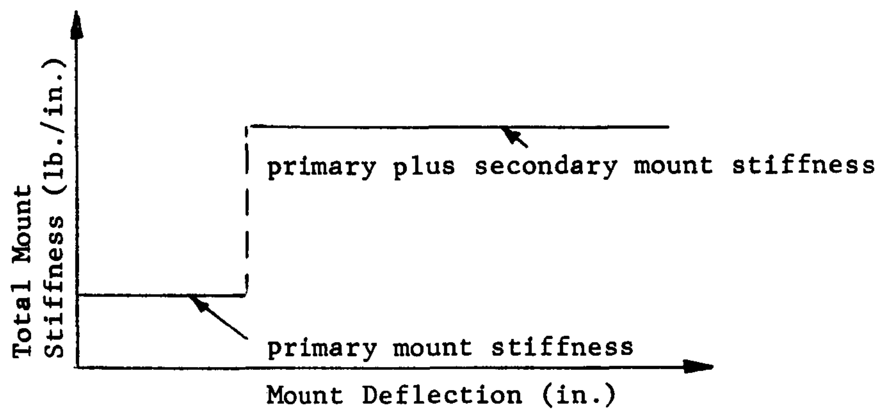
2. The primary mount has been designed to be operative over a range of vertical accelerations from zero to 1.2 g. During launch, when vertical accelerations exceed 1.2 g, the compressor engages a secondary mount system consisting of two cantilever beam springs. The vertical natural frequency of the secondary mount is 14.3 hz. During the time when the compressor is supported by the secondary mount system, the dynamic force which will be transmitted to the IU panel is calculated to be $\pm$ 4.8 pounds.

3. The primary mount system has been designed such that the coil springs will always be in tension regardless of the acceleration g-level.

4. Lateral position of the compressor is maintained by the primary mount system during normal zero to 1-g operation. However, since the lateral spring rate of the primary system is quite low, three soft bumpers, equally spaced around the compressor, are used to limit excessive lateral movement of the compressor as might occur during launch

or mission maneuvers. Since there are no dynamic lateral forces produced by the compressor, there will be no force transmission to the IU panel during the time when the compressor is in contact with one of the bumpers.

Recent tests by MSFC have identified a possible problem area in the presently designed mount system. The secondary cantilever springs introduce a step change in mount stiffness at a particular value of mount deflection as shown below.




Mount Stiffness Versus Mount Deflection

As a result of this discontinuity in the stiffness characteristic, a 60 hz intermittent contact can occur between the compressor casing and the secondary cantilever springs. It can be shown that under ideal conditions this intermittent contact condition would exist only for a small range of acceleration, from 1.2 g to approximately 1.5 g. However, in practice the contacts would probably occur over a slightly larger range of acceleration due to manufacturing and set-up tolerances associated with the secondary mount system. As presently designed, the secondary mount system permits metal-to-metal contact between the compressor casing and the cantilever springs. Hence, during the period of intermittent contact (that is, when acceleration is between 1.2 and, say, 2 g), a considerable amount of noise will be produced by the secondary mount system. Aside from a human factors consideration (which would probably not be important in the ultimate mission, but which should be considered with regard to the large amount of laboratory testing prior to the mission), the noise, per se, should not be a problem. However, the points of intermittent contact are potential points of brinelling and hence should be evaluated from this respect. In addition, the intermittent contacts are, in effect, small repeated impacts imposed on the compressor and hence may affect compressor performance and reliability. These possibilities should likewise be checked.

In spite of the possibilities identified above, it is believed that the concept of the present compressor mounting scheme is a practical one. It is felt that the potential problem areas can be acceptably resolved by revised mechanical design. With respect to the presently designed mount system, preliminary testing has revealed the following needed improvements:

1. A minimum of three, rather than the present two, cantilever springs should be used to assure static stability of the secondary mount system.

2. The frequency of the secondary mount system should be reduced from 14.3 hz to a maximum of 10 hz. This will reduce the transmitted force from $\pm$ 4.8 pounds to a maximum of $\pm$ 2.3 pounds during high-g acceleration.

3. A sound and impact "deadening" material should be used at the contact points between the compressor casing and the secondary cantilever springs.

4. Detail design of both the primary and secondary spring systems should be improved to permit more accurate set-up and alignment of the mount system.

5. Although not required for the ultimate mission, design of the mount system should be revised to permit centrifuge acceleration testing in a 1-g field with the compressor mounted in a horizontal position.

## Compressor Mechanical Design

A complete set of Parts Lists for the Resonant Compressor assembly and for the Gas Supply System assembly are included in Appendix D of this report for reference purposes. The following paragraphs describe the mechanical design of the compressor in more detail.

The piston and the cylinder for the compressor are made of carbon graphite and Ni-Resist iron respectively. The reasons for this combination are as follows:

1. Each material has high dimensional stability

2. Coefficients of expansion of the two materials are closely matched

3. The carbon has low mass density (about the same as magnesium) which is quite important from a dynamics and resonant-spring stress standpoint

4. The two materials have very good rubbing compatability

5. Both materials are corrosion resistant.

Photographs of the piston and cylinder are shown in Figure 17.

The measured weights of the reciprocating compressor parts are as follows:

| | | |
|---|---|---|
| 1. | Piston (dw'g. 141 B07) | 0.231 lb. |
| 2. | Solenoid plunger (dw'g. D-1253) | 0.250 |
| 3. | Connecting rod (dw'g. 141 C08, Pts. 1 & 2) | 0.084 |

| | |
|---|---|
| 4. Ring (dw'g. 141 B51) | 0.047 |
| 5. Hardware<br>One 10-32 screw (pt. 35)<br>Four 10-32 screws (pt. 34)<br>Two 8-32 screws (pt. 40) | 0.037 |
| 6. Moving platen (i.e., thickened end plate) of the resonant spring assembly (dw'g. 141 D1) | 0.140 |
| Total reciprocating weight | 0.789 lb. |

A portion of the total weight of the four U-spring sections of the resonant spring assembly must also be considered as part of the reciprocating weight. However, this weight is automatically factored into the calculation procedure for producing the spring design map shown in Figure 8. The weight parameter of Figure 8 is therefore total reciprocating weight over and above the effective weight of the U-spring sections themselves.

The compressor is mounted inside a hermetically sealable aluminum container (the present containers have also been equipped with O-ring-sealed bolt flanges for ease of access during the evaluation period). The inside of the container is vented to compressor inlet pressure. All piping and electrical connections are made to fittings located on the outside of the container. Figure 18 is a photograph of the two sections of the container.

Three liquid-cooled heat exchangers are located inside the compressor container. Two of the exchangers are gas-to-liquid types used for temperature conditioning of the nitrogen flow into and out-of the compressor. The third exchanger is simply a section of the liquid coolant line between the two gas conditioners which has been wrapped around and potted to the outside surface of the drive solenoid coil. This exchanger removes the 18 or so watts of $i^2R$ loss in the solenoid coil.

The gas-to-liquid heat exchangers which condition the nitrogen flow into and out-of the compressor are aluminum finned-tube types. Nitrogen flows through an OD spiral passage formed by the fins and the outer casing of the exchanger. The liquid coolant flows through a straight section of tubing to which the fins are attached. Coolant temperature into the compressor container will be $59 \pm 1^{\circ}F$. Calculated temperature rise of the coolant after it passes through the two gas coolers and the solenoid cooling loop is $7^{\circ}F$. Flow rate of the coolant was taken as 200 pounds per hour. Nitrogen pressure drop through each gas cooler was calculated to be 0.23 psi at 0.5 SCFM. Measured pressure drop at the same flow was found to be 0.2 psi for the coolers supplied with the first delivery compressor. Measured pressure drop was considerably higher for the second delivery compressor, even though the cooler design was the same. The reason for the higher pressure drop was traced to incorrect manufacture of the second set of coolers. This condition was discovered after the second container assembly had been completed, and hence was not corrected due to schedule and funding limitations. The two gas coolers can be seen in Figure 19 as the two vertical tubes attached to the bottom section of the container.

## IV. PROCUREMENT OF THE GAS MAKE-UP SUBSYSTEM

The Gas Make-Up Subsystem, as shown schematically in Figure 1, was procured as an integral package from Aerodyne Controls Corporation of Farmingdale, Long Island. The complete subsystem is designated as Aerodyne Controls Corporation P/N 670-1-000 and is shown on Aerodyne drawing 670-1-000B. The subsystem, as originally proposed by Aerodyne, is described in Aerodyne Engineering Proposal No. 670 (this proposal also describes the differential pressure regulator as procured from Aerodyne for the Recirculation Subsystem).

The Gas Make-Up Subsystem contains four major groups of parts as follows:

1. Gas reservoir (or storage vessel)

2. Manifold assembly, containing

    a) one flow limiting orifice in the reservoir charging line

    b) two identical charging valves connected in series, but physically displaced 90 degrees from one another

    c) one pressure relief valve

    d) one rupture disk

    e) one pressure switch

    f) one filter assembly

    g) one flow limiting orifice in the make-up line to the absolute pressure regulator

3. Absolute pressure regulator

4. Absolute pressure relief valve.

### Gas Reservoir

The gas reservoir is a sphere of AMS 6350 alloy steel with a single AND-10050-16 port and is designed for normal working pressures of 3000 to

3300 psig. Burst pressure is in excess of 6000 psig. The reservoir has been subjected to a hydrostatic proof pressure of 4500 psig for 5 minutes with no evidence of yielding or damage.

The reservoir has a volume of approximately 900 cubic inches, is approximately 12.7 inches in diameter, and weighs 22 pounds. The internal and external surfaces of the vessel are electroless nickel plated for corrosion resistance.

Manifold Assembly (P/N 671-1-000A)

The manifold of the assembly is made from 2024-T4 aluminum alloy anodized per MIL-A-8625A, Type I, for corrosion resistance. The manifold contains an AND-10050-4 charging port in which is located an 8 mil diameter flow limiting orifice. The charging-flow limiting orifice (1) minimizes pressure shock loads on the manifold components, (2) prevents a too-rapid build-up of pressure with its consequent large rise in temperature within the reservoir, and (3) permits the use of a smaller size pressure relief valve for reservoir protection. The manifold also contains two charging valves, a relief valve and burst disk for reservoir protection, a pressure switch for storage vessel pressure indication, a threaded boss for attachment to the reservoir, and, in the outlet port, a filter and flow limiting orifice. The manifold is stressed for pressures in excess of 6000 psi.

Deviations from Specification

During the design and test phases of the Gas Make-Up Subsystem procurement, Aerodyne requested permission to make several deviations from the subsystem specifications. These deviations, which were subsequently approved by MTI (and, where necessary, by MSFC) are listed below.

1. A request was made to reduce the inlet surface area of the high pressure filter (item L of Figure 1) from 3.0 square inches to 0.11 square inches. For the filter element itself, a Millipore type RAWP01300 was proposed, to be held between two photo-etched stainless steel prefilters. The primary filter would have pores

of 1.2 $\pm$ 0.3 microns and a minimum volume of 110 x $10^{-6}$ cubic inches.

The main area of concern with the filter is that of clogging due to oil contamination. The reservoir, when charged to 3100 psia, will hold approximately 8.2 pounds of nitrogen. If the nitrogen, as charged, contains 0.3 parts per million by weight of contaminating oils, then the maximum volume of oil in the reservoir will be 85 x $10^{-6}$ cubic inches (assuming a specific gravity of 0.8 for the oil). Therefore, even if all the oil in the reservoir were to be captured in the filter, the filter would not be completely clogged.

The question here, then, is one of original contamination level of the nitrogen as charged into the reservoir. This question should be pursued, perhaps by test, to determine if oil clogging will be a problem.

2. A request was made by Aerodyne to substitute an Aero Mechanisms Incorporated Model 6455, S/N 118, pressure switch for the originally proposed switch. Although the originally proposed switch met the specified electrical rating of 1 amp into an inductive load at 28 VDC, it was found on actual placement of the purchase order that the switch was no longer being produced, and would have to be manufactured as a special at considerably higher cost and with a 4 month delivery.

   After careful searching, only one other switch was found of reasonable size, cost, and delivery which would satisfy all requirements except for electrical rating, this being the Aero Mechanisms switch. The electrical rating for this switch is 0.2 amps into a resistive load at 28 VDC. Since the switch is only to be used for reservoir pressure verification, it was felt that the lower ampere rating would still be quite adequate, at least for the development model of the Make-Up Subsystem.

3. A request was made to permit a significant deviation on the reseating pressure of the reservoir pressure relief valve. The relief valve

requirement specified a maximum cracking pressure of 3300 psia and a minimum reseat pressure of 3150 psia. The relief valve as finally redesigned has a cracking pressure of 3275 $\pm$ 25 psia and a reseat pressure of 600 $\pm$ 50 psia. The following excerpts from an Aerodyne letter to MTI describe the reason for the deviation request.

> As originally designed, the relief valve was tested and found to meet the letter of the requirements, that is, cracking pressure was 3280 $\pm$ 20 psia and reseat pressure was 3180 $\pm$ 30 psia. However, this relief valve, as originally designed, did not meet what Aerodyne Engineering considered to be the intended function of the relief valve; the capability of providing adequate protection for the reservoir against overpressurization. This relief valve, although it met specification pressure requirements, did not flow more than 10 SCCM at 3400 psia and even at pressures of 3600 psia did not flow more than 100 SCCM. Since the flow rate into the manifold assembly and reservoir during system charging greatly exceeded the relief valve flow capability, the system was possibly subject to overpressurization during charging. The result could have been rupture of the burst disc assembly included in the manifold as a protection against relief valve failure and reservoir overpressurization. This burst disc is not easily replaced and rupture of it would probably necessitate a lengthy recleaning process of the manifold assembly.
>
> With the decision that the originally designed relief valve was inadequate, Aerodyne Engineering instituted a program for development of a relief valve capable of adequately protecting the system from overpressurization. Two design solutions were possible. The first solution was a design sufficiently large to be capable of high relief valve flow. This design could not be incorporated in the manifold assembly as originally desired and, because of the large, heavy spring necessary for proper performance, caused a severe penalty in package and weight (in the order of 0.75 lbs). In addition, provisions had to be made for externally mounting this design.
>
> The second design solution was what is termed a differential area type relief valve. This type of relief valve is characterized by a very sharp cracking pressure with excellent repeatability within a minimum size and weight package. Due to the differential area in the poppet-piston, there must be a substantial decrease in pressure before this type of relief valve will reseat.

In view of the fact that overpressurization is likely to occur only during reservoir charging operations, where care can prevent such occurrences, and realizing that this design offers excellent reservoir protection, is of minimum size and weight, and can be built into the manifold assembly, it is Aerodyne's engineering opinion that this second design solution is considerably better than the first design solution. As a consequence of this, a differential area piston type relief valve was designed and developed for reservoir protection in the Gas Makeup Supply System.

During a telephone conversation between Dr. Peter Curwen of MTI and Mr. L. P. Tetrault of Aerodyne Controls, on December 13, 1965, the failure of the original design to meet its intended purpose and preliminary results of testing on the new differential area type relief valve were discussed. Since the redeveloped relief valve had exceptionally high flow capability, had a very sharp, repeatable cracking pressure of 3275 ± 25 psia, had zero leakage at pressures to 3200 psia, was extremely small in size and weight, and could be built into the manifold assembly, Aerodyne considered this relief valve far superior to the originally designed relief valve.

A further consideration to be noted is that although the improved performance was made possible only by a substantial decrease in relief valve reseat pressure (600 ± 50 psia), the relief valve is only a secondary component, that is, it is not necessary to system operation. Since reservoir overpressurization is likely to occur only during system charging, the low reseat value means merely the loss of several pounds of nitrogen gas.

## Acceptance Test

The Gas Make-Up Subsystem was subjected to the following tests by Aerodyne Controls Corporation:

1. Reservoir proof pressure

2. Actuation and de-actuation pressure of the pressure switch

3. Cracking and reseat pressure of the reservoir pressure relief valve

4. Cracking and reseat pressure of the absolute pressure relief valve

5. Flow calibration of the flow limiting orifice (make-up flow line to absolute pressure regulator)

6. Cracking pressure of the charging valves

7. Regulation characteristics of the absolute pressure regulator

8. Total system external leakage

9. Total system cleanliness.

The Aerodyne Acceptance Test Report (A.C.C. Document No. TR-670) is included in Appendix C of this report for reference purposes. The results of all tests were satisfactory.

## V. TEST EVALUATION

### First Delivery Compressor

The first resonant compressor was assembled and tested during August and September, 1965. Very little trouble was encountered with the assembly and start-up process. The compressor was instrumented with five capacitance probes, four to monitor performance of the piston bearing and one to measure deflection of a small diaphragm in response to instantaneous cylinder pressure. Two thermocouples were also installed, one to measure cylinder head temperature and the other to measure surface temperature of the drive solenoid coil. A strain gage was cemented to one of the U-spring sections close to its fixed end to provide a piston displacement signal.

Figure 19 is an overall view of the compressor test area. The first compressor assembly, with the top section of the aluminum container removed, is shown with the various instrumentation lines connected to the readout equipment. Testing was accomplished with the inlet fitting of the inlet gas cooler open to atmospheric air. Compressor discharge was connected, through the discharge gas cooler, to a 17 cubic inch reservoir at which point discharge pressure was measured. From the reservoir the compressed air passed through a flow meter and then through a throttle valve which was used to simulate various load conditions. After flowing through the throttle valve, the air was vented back to atmospheric conditions.

Figures 20 through 24 show measured performance characteristics of the first compressor at 5, 10, 15, 17.5, and 20 psi pressure rise. During these tests the unit was driven from a constant voltage 60 hz power line with a diode in series with the drive solenoid so as to obtain a 60 hz fundamental drive force from the solenoid. Figures 20 through 24 show oscilloscope pictures of the:

1. compressor p-V diagram

2. simultaneous traces of cylinder pressure and piston displacement versus time

3. simultaneous traces of solenoid current and piston displacement versus time

4. orbit traces of the piston inlet (upper) and discharge (lower) bearings.

The p-V diagram traces show very nicely the nature of the inlet and discharge intervals of the cycle. Note that the diagrams agree quite well with the expected diagrams as shown in Figures 4 and 5, although there are some minor differences in the details of the pressure variations during the inlet and discharge intervals. Maximum discharge-valve pressure drop is between 1 and 2 psi which agrees well with the calculated pressure drop.

The compressor was tested for a total of 115 hours at MTI, 98 hours of which was accumulated during a continuous run at design point conditions. During these 115 hours of operation, $2.48 \times 10^7$ stress cycles were accumulated on the resonant spring and discharge valve. Since the "knee" of the fatigue curve for both the valve and spring material is reached in about $1.0 \times 10^6$ cycles, it was concluded that the compressor design was satisfactory from a short-term fatigue and fretting standpoint.

Subsequent to the 115 hours of operation, the compressor was completely dismantled and carefully examined for evidence of rubbing, wear, fretting, or other damage. None was found.

The unit was reassembled, permanent instrumentation installed, and performance rechecked. Figure 25 shows the final flow versus pressure rise calibration, as a function of drive solenoid voltage, taken after three hours of continuous operation at design point conditions to achieve thermal equilibrium. Subsequent to this final calibration, the compressor was shipped to MSFC.

Spare Compressor

The spare compressor was assembled, instrumented, and tested in the same manner as the first delivery compressor. A total of 162 hours of operation was accumulated on this unit at MTI, 153 hours of which was obtained during a continuous run. Figures 26, 27 and 28 show compressor performance conditions at the 9th, 120th, and 150th hour, respectively, of the 153 hour continuous run. It is seen from these figures that there was no detectable change in performance except for an increase in flow between the 9th and 120th hour due to an increase in solenoid voltage at the 94th hour, and a slight contraction of the discharge (lower) bearing orbit, also due to the voltage change (discharge pressure was held constant during the voltage increase by re-adjustment of the throttle valve).

The spare compressor is definitely a more efficient machine than the first delivery unit, even with the increased pressure drop in the two gas coolers. This can be seen by comparing Figures 25 and 29, these being the flow versus pressure rise calibrations as a function of solenoid voltage for the two machines. It is seen that the spare machine can deliver any given combination of flow and pressure at a lower solenoid voltage than can the first delivery machine. From the constant efficiency lines superimposed on Figure 29, it is seen that the spare compressor can deliver design-point flow at an overall efficiency of better than 33 percent. The performance curves of Figures 25 and 29 were obtained with compressor inlet and discharge flow passing through the gas coolers. Hence the efficiencies shown on Figure 29 are not, in a strict sense, a true measure of compressor efficiency since the effects of cooler pressure drops are included in these efficiency values. A more accurate representation of performance of the spare compressor is shown by Figure 30, which is the flow versus pressure rise characteristic when the gas coolers are by-passed. From Figure 30 it is seen that the true design-point efficiency of the spare compressor is close to 60 percent. It is quite probable that if the gas coolers for the spare machine had been properly fabricated (which would have reduced the pressure drop through each cooler to about 0.2 psi) the efficiency of the spare unit with coolers included would have been about 50 percent.

There are several reasons why the spare compressor could have a better efficiency than the first delivery unit.

1. The spare unit is believed to be better tuned, mechanically, for 60 hz resonant operation. The reciprocating assembly for this unit is slightly lighter than the first machine.

2. It is believed that the average solenoid air gap is slightly smaller for the spare unit. This would result in less solenoid current, and hence less $i^2R$ loss, for the same drive force levels.

3. Piston clearances may be smaller in the spare unit. This could considerably increase efficiency.

The significant point from Figure 30 is that very good overall efficiencies can be obtained from the gas-lubricated resonant compressor concept. With further optimization studies and better understanding of the design principles, it appears that overall efficiencies between 70 and 80 percent can be obtained.

Subsequent to completion of testing, the spare compressor and the Aerodyne Gas Make-Up Subsystem were shipped to MSFC. Figures 31 and 32 show a close-up view of the spare compressor and the Gas Make-Up Subsystem respectively.

## Required System Modification

There is one important limitation of the present compressors. As compressor $\Delta p$ is increased, total piston stroke likewise increases for any given value of solenoid voltage. This is illustrated in Figure 33 which shows relative piston stroke versus pressure rise for the spare compressor as a function of solenoid voltage. Above 75 volts, it is possible to bottom the solenoid plunger on the solenoid stator as low-flow, high $\Delta p$ conditions are obtained.

From a system standpoint, excessive build-up of piston amplitude can be prevented by a simple change in regulation concept which will prevent the build-up of high $\Delta p$'s. From the Gas Supply System schematic shown in Figure 1, it is seen that the platform $\Delta p$ regulator consists of a variable orifice in series with the platform load. This type of regulator tends to maintain constant compressor flow, and dissipates any excess compressor pressure rise across the variable orifice. This mode of regulation was chosen in anticipation that the resonant compressor flow-pressure characteristics would be relatively flat in the vicinity of the design-point conditions. However, it is seen from Figures 25 and 29 that the flow-pressure characteristics are actually quite vertical in the design-point region. Consequently, the platform $\Delta p$ regulator should be changed from a constant flow, variable pressure type of regulator to a constant pressure, variable flow unit which will bypass excess compressor flow around the platform. This type of regulation is basically check-valve regulation.

The check-valve regulator, since it limits compressor pressure rise, will prevent excessive build-up of piston amplitude. This type of regulation has already been demonstrated by MSFC, using the first delivery compressor, as being very effective.

## CONCLUSIONS AND RECOMMENDATIONS

The primary consideration for successful development of a Gas Supply System for Hermetically Sealed Platforms is the availability of a compressor to meet the following requirements:

1. long, maintenance-free life (in excess of 10,000 hours continuous operation),

2. hermetically sealable construction,

3. truly non-contaminating operation (less than one part per million by weight of condensable hydrocarbon content),

4. ability to operate during launch,

5. ability to operate over an ambient temperature range of 0 to 60 degrees C,

6. minimum possible drive power,

7. minimum weight and size consistent with the above objectives.

A Resonant Piston Compressor utilizing a gas-lubricated piston was proposed as a unique approach to satisfying the above requirements. Two such compressors have been built and successfully tested at MTI. These compressors are now being evaluated by NASA-MSFC, with one unit having presently accumulated over 2,700 hours of trouble-free operation and no degradation of performance. The second unit is also accumulating operating hours, at a somewhat slower rate, and has likewise given trouble-free operation to date.

Perhaps the most striking attribute of the resonant compressor concept, from the standpoint of ultimate space system applications, is its high overall efficiency. One of the units presently being tested has shown a basic overall efficiency of almost 60 percent at the required design-point conditions. Maximum efficiencies measured with this unit (at higher than required pressures) were almost 75 percent. These efficiencies are very high relative to what can presently be obtained with conventional, high-speed, low-flow compressors. With development of better understanding of the operating principles of the resonant compressor, and by putting more emphasis on minimizing internal leakages, it appears that efficiencies in the range of 80 to 85 percent may be attainable. The importance of high component efficiency with respect to minimizing space vehicle power supply weight and size is well known.

## Recommendations

Several noteworthy improvements are now recognized as being feasible and desirable with respect to design of the resonant compressor.

1. It is evident from Figures 19, 31 and 32, that the envelope size of the compressor is determined by the resonant spring assembly. A 66 percent reduction in envelope size now appears feasible using a special coil spring design in place of the flat U-shaped springs presently being used. In addition to the large reduction in envelope volume, some reduction in weight will also be realized as a result of the smaller container size.

2. Further weight reduction appears feasible by substituting aluminum wire for the copper wire presently used in the drive solenoid coil. A reduction of about five pounds (i.e., about 12 percent of the present compressor weight) is possible, although a slight increase in $i^2R$ coil losses will probably be incurred.

3. The compressor has already been successfully tested by MSFC in a 10-g acceleration field with the acceleration vector parallel to the piston displacement. Performance was virtually unchanged under this condition. However, with the acceleration vector

perpendicular to the piston displacement, compressor performance starts to drop-off between 2 and 3 g's. This is undoubtedly due to sliding contact between the piston and cylinder resulting from the large bearing reaction forces produced by the transverse g-loads on the overhung solenoid plunger.

To overcome this condition, it would be possible to augment the self-acting piston bearings with a small amount of hydrostatic bearing action using compressor discharge for bearing pressurization. Although this will decrease the net compressor efficiency, the bearing flow can be minimized so as to provide just the additional load capacity required to support the specified g-level over and above 2 g's.

Some additional transverse load capacity may also be realized by virtue of the coil spring design recommended above in place of the present U-shaped springs.

4. Additional study of the compressor mounting system should be made taking into account the effect of transmittal force level on the IU mounting panel. A realistic mounting system specification should be defined, and commercially available mounts and mounting systems evaluated in terms of the specification. If commercially available mounts can not be made, or modified, to satisfy the specification, then the presently designed mounting system should be further improved as described on page 22.

5. The overall mechanical configuration of the compressor should now be reviewed and modified as required to satisfy all applicable Saturn V component design, fabrication, material, reliability, and cleanliness specifications.

6. Optimization of the compressor drive circuit and drive solenoid should be undertaken. To this end, a better analytical understanding of the drive solenoid operating principles must be obtained, together with specific design procedures and equations.

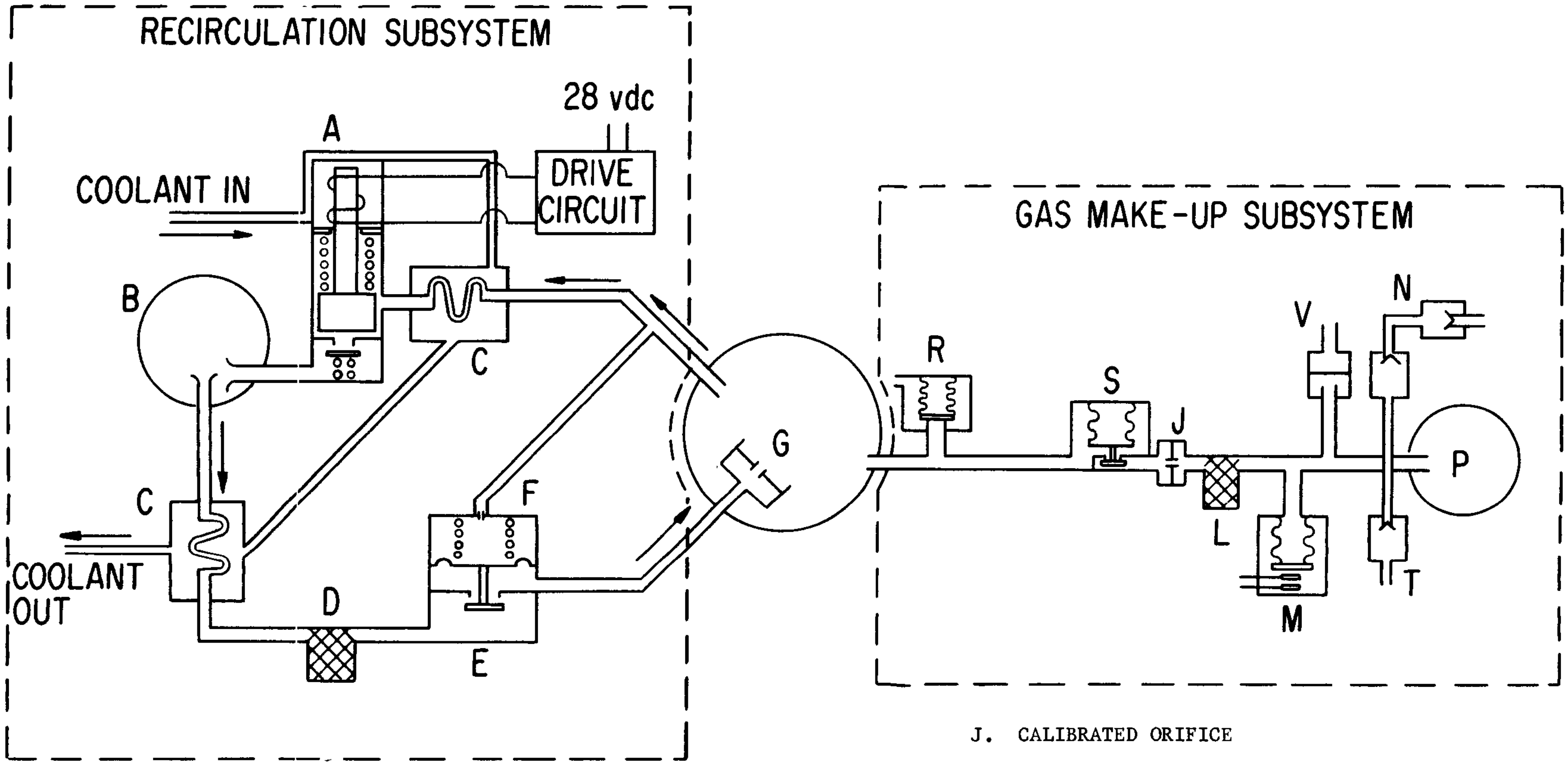
7. The operating principle of the resonant compressor appears to be quite adaptable to automatic $\Delta p$ control. By using closed-loop feedback control techniques to continuously and automatically match compressor output to platform $\Delta p$ requirements, the overall efficiency of the Gas Supply System can be significantly improved. This approach would eliminate the need for the $\Delta p$ regulator shown in Figure 1, as well as a voltage regulator for the compressor drive circuit. Both of these components, and particularly the $\Delta p$ regulator, are sources of power loss and hence are detrimental to overall system efficiency.

   Perhaps the most significant aspect of feedback control of platform $\Delta p$ is the ability to preprogram, or to randomly select, the desired value of platform $\Delta p$ using an on-board computer, telemetry, etc. The ability to power-down the compressor when high platform $\Delta p$ is not required, by changing the $\Delta p$ set-point value, would result in very significant power savings.

8. The use of an inlet valve instead of an inlet port for the compressor should be reviewed from a reliability standpoint. The present inlet port is certainly the utmost in simplicity and reliability as far as valve mechanism is concerned. However, operation of the port requires achievement of a certain minimum piston displacement before inlet flow can begin. Operation of a valved inlet, however, is dependent on cylinder pressure rather than a specific piston displacement. Hence, if for some reason piston displacement should become constrained or limited, such as was perhaps the case with the transverse centrifuge tests performed by MSFC, the compressor would continue to produce some output with a valved inlet, as opposed to a very rapid drop-off in output with the ported inlet.

Use of a valved discharge in the present compressors has demonstrated that reliable valves can be obtained. Perhaps the optimum inlet arrangement would be a combination of the present inlet ports supplemented by a small inlet valve.

FIGURES




A. RESONANT PISTON COMPRESSOR

B. SURGE CHAMBER

C. GAS HEAT EXCHANGER

D. LOW PRESSURE FILTER

E. LOW PRESSURE REGULATOR

F. FLOW RESTRICTOR

G. INERTIAL PLATFORM

J. CALIBRATED ORIFICE

L. HIGH PRESSURE FILTER

M. PRESSURE SWITCH - 3000 PSI CONFIRMATION

N. TWO CHARGING VALVES - RIGHT ANGLES

P. RESERVOIR - 3000 PSI

R. RELIEF VALVE

S. ABSOLUTE PRESSURE REGULATOR

T. RELIEF VALVE - RESERVOIR

V. BURST DISK - RESERVOIR

Fig. 1 - Schematic Diagram of Gas Supply System

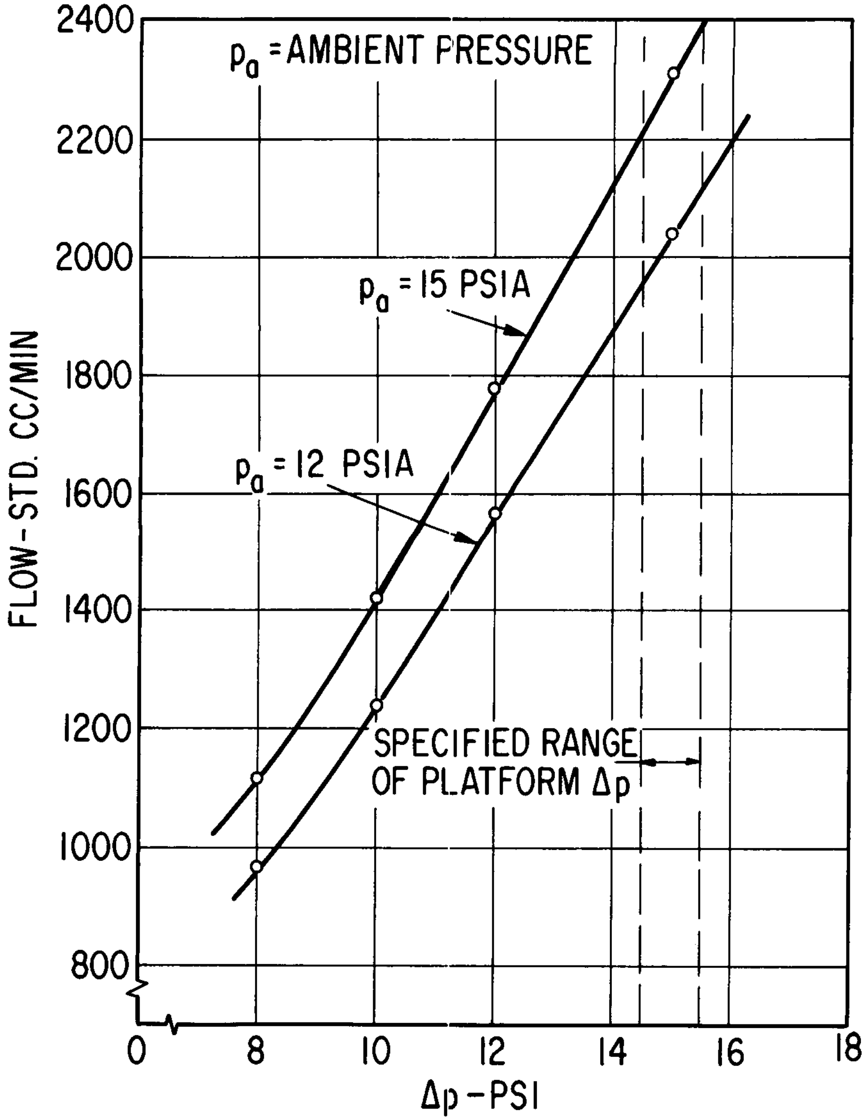



Fig. 2 - Nitrogen Flow Versus $\Delta p$ for One AMAB-3 Accelerometer

MTI-1740

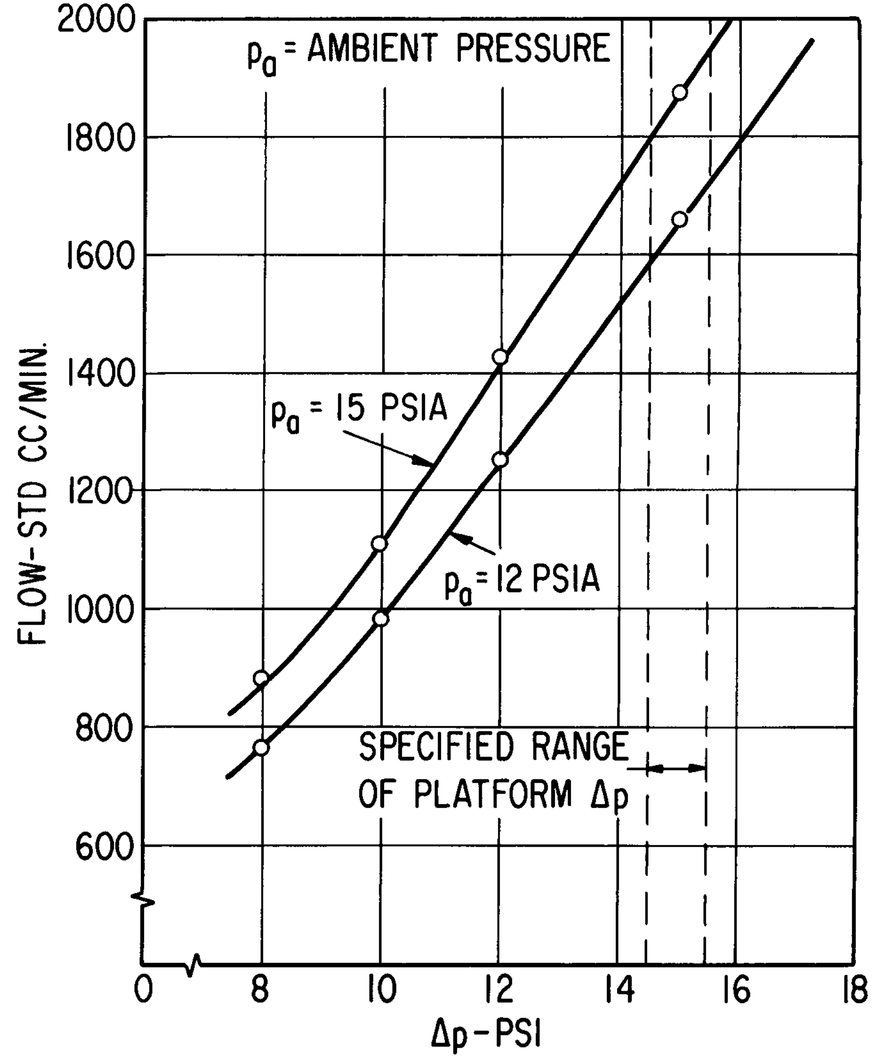



Fig. 3 - Nitrogen Flow Versus $\Delta p$ for One AB-5 Gyro

MTI-1741

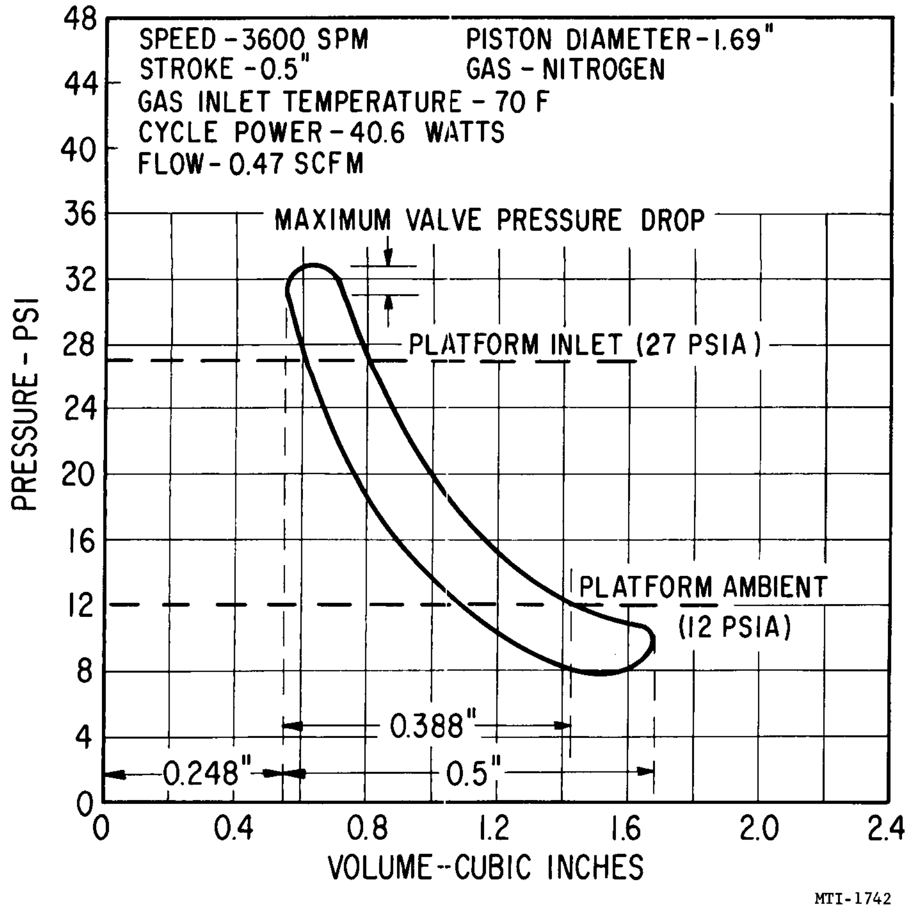



Fig. 4 - Resonant Compressor p-V Diagram for 12 PSIA Inlet Pressure

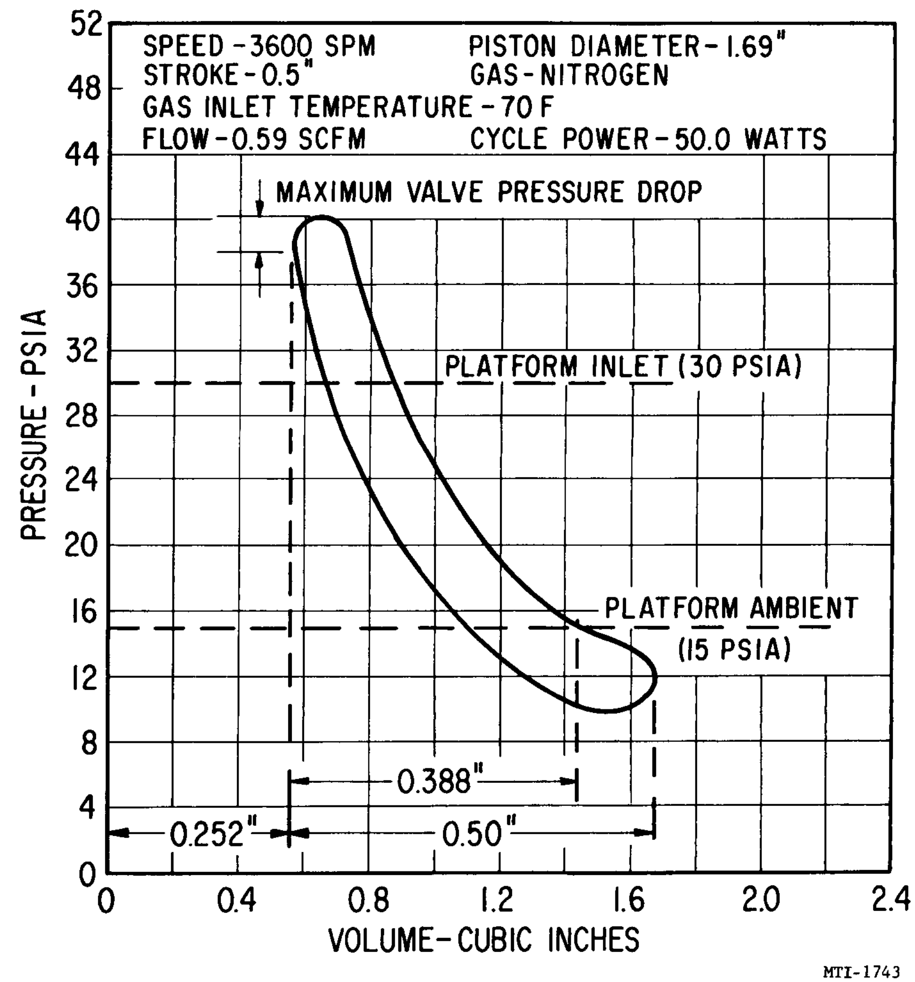



Fig. 5 - Resonant Compressor p-V Diagram for 15 PSIA Inlet Pressure

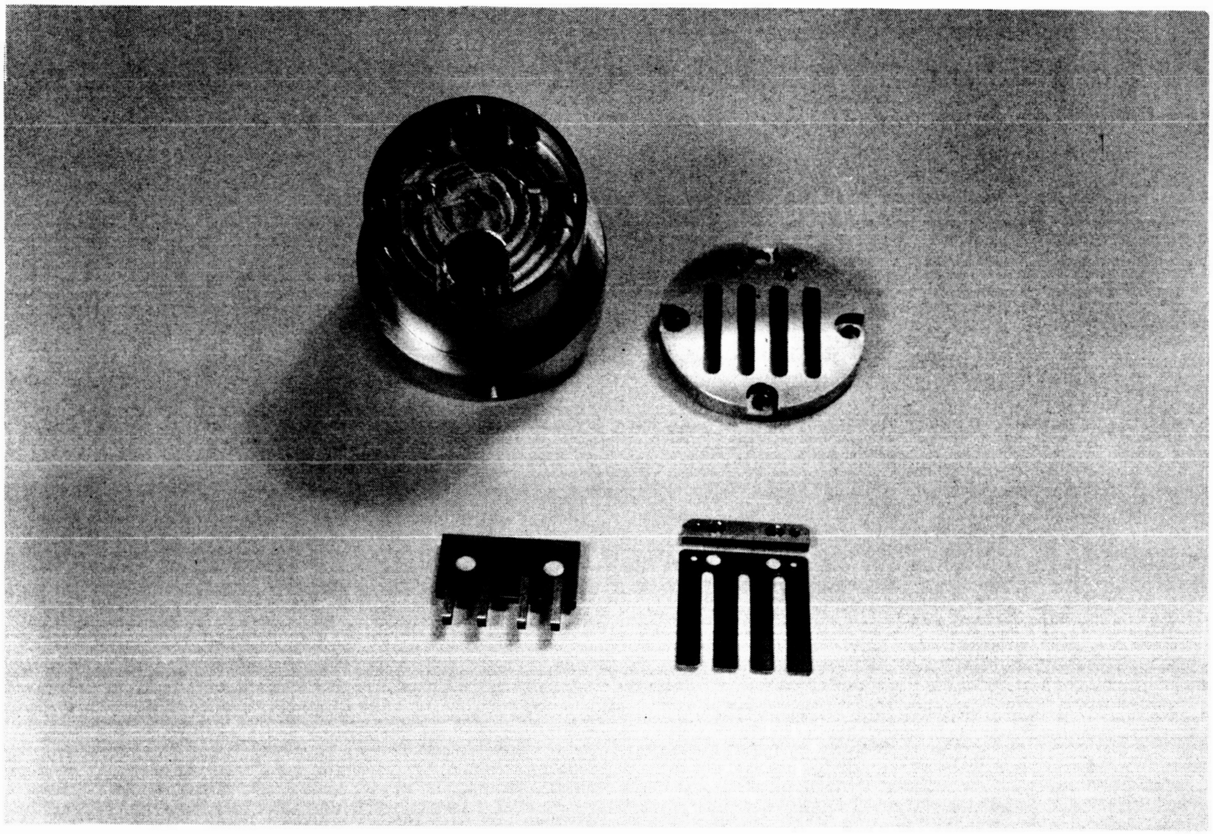


Fig. 6 - Resonant Compressor Discharge Valve Components

MTI-1744

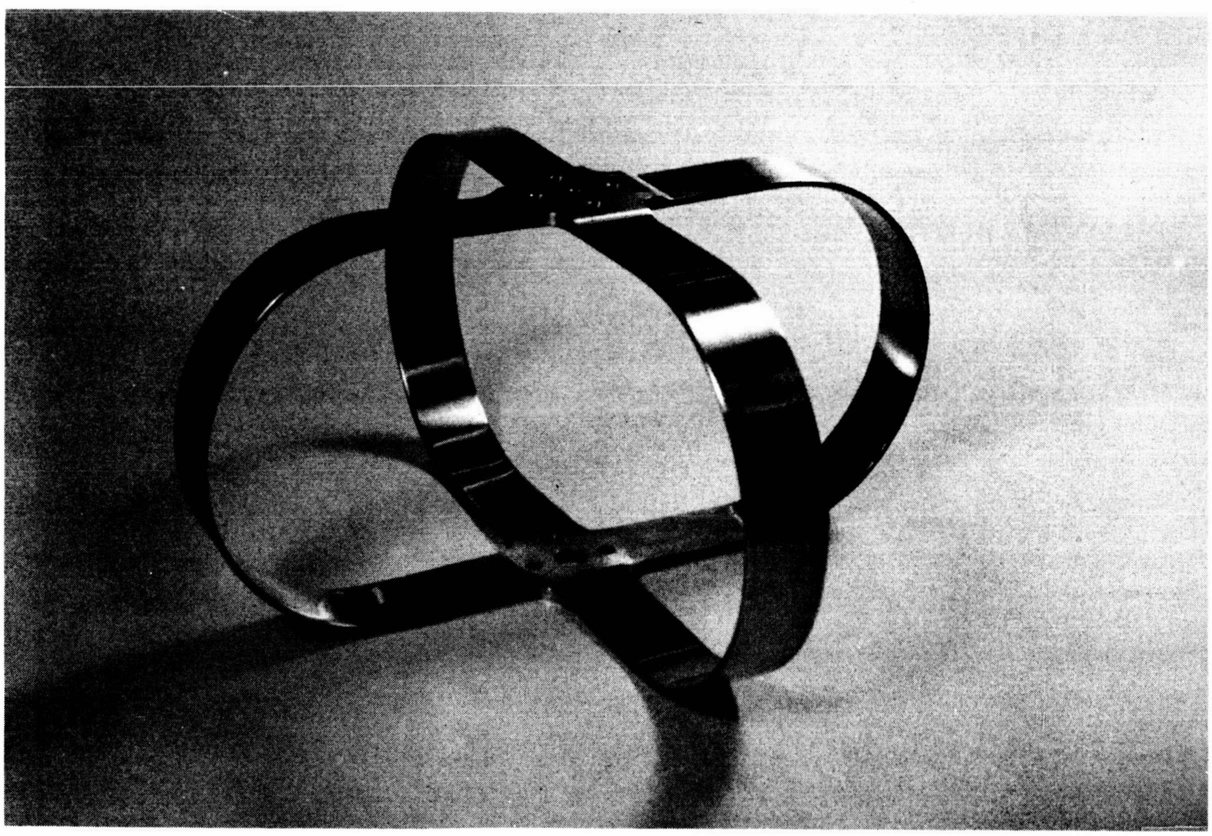


Fig. 7 - Resonant Spring Assembly for Resonant Compressor

MTI-1745

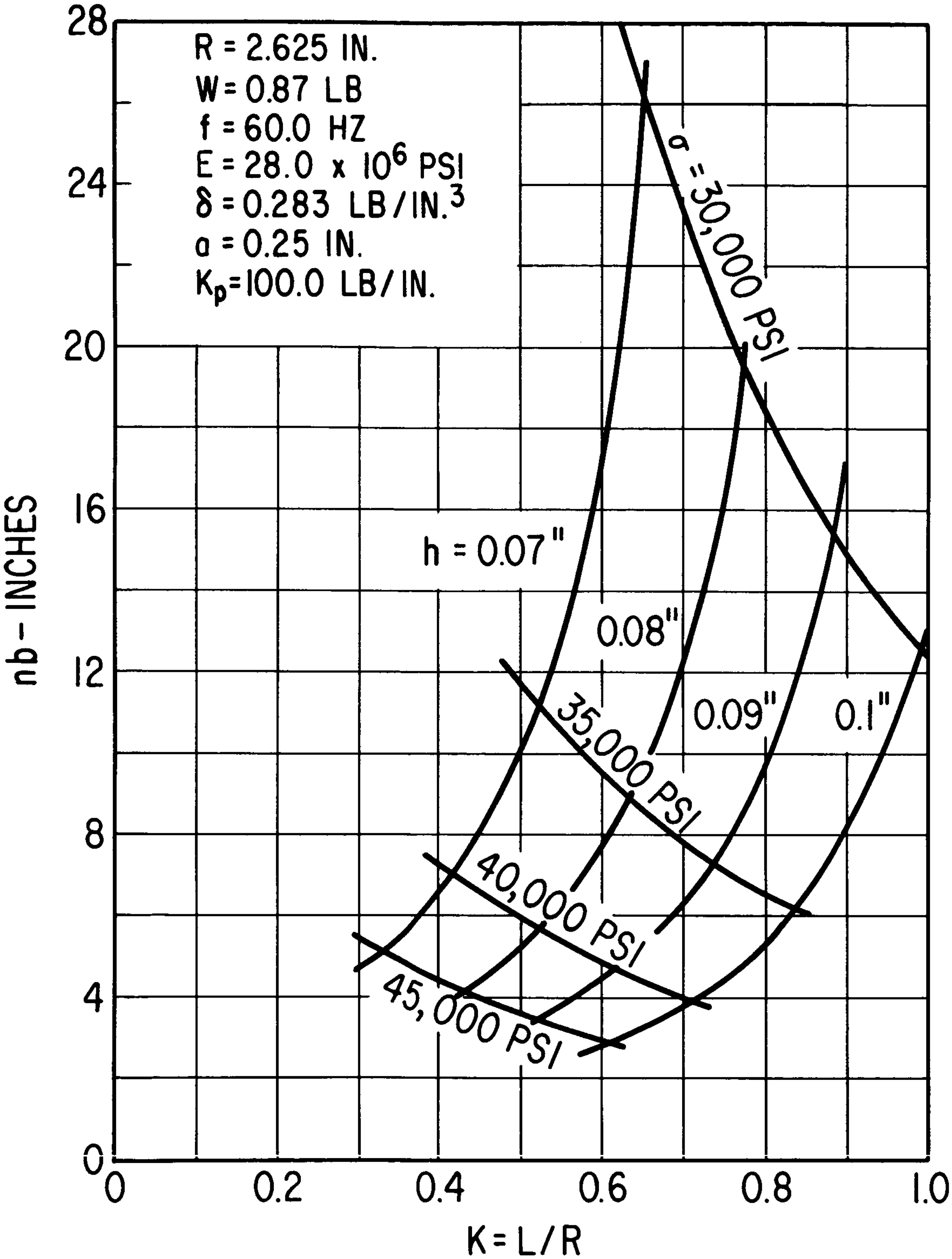
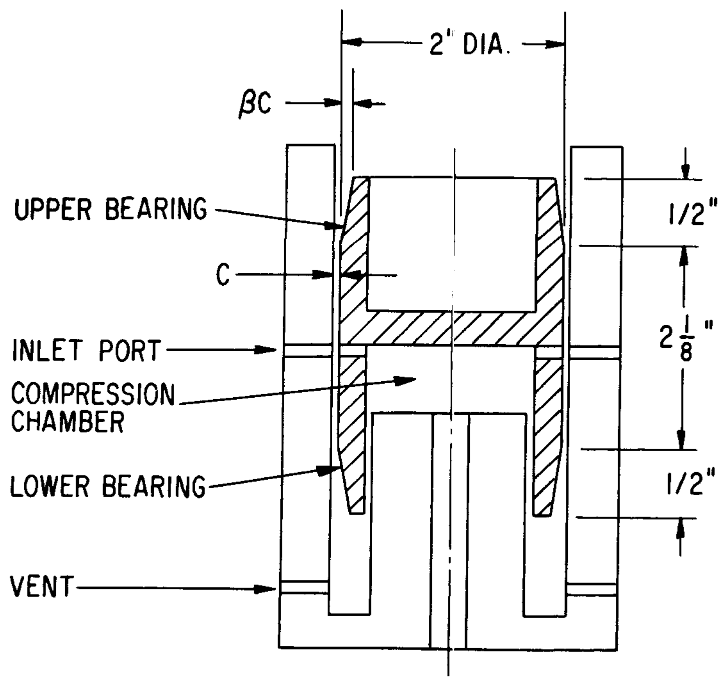
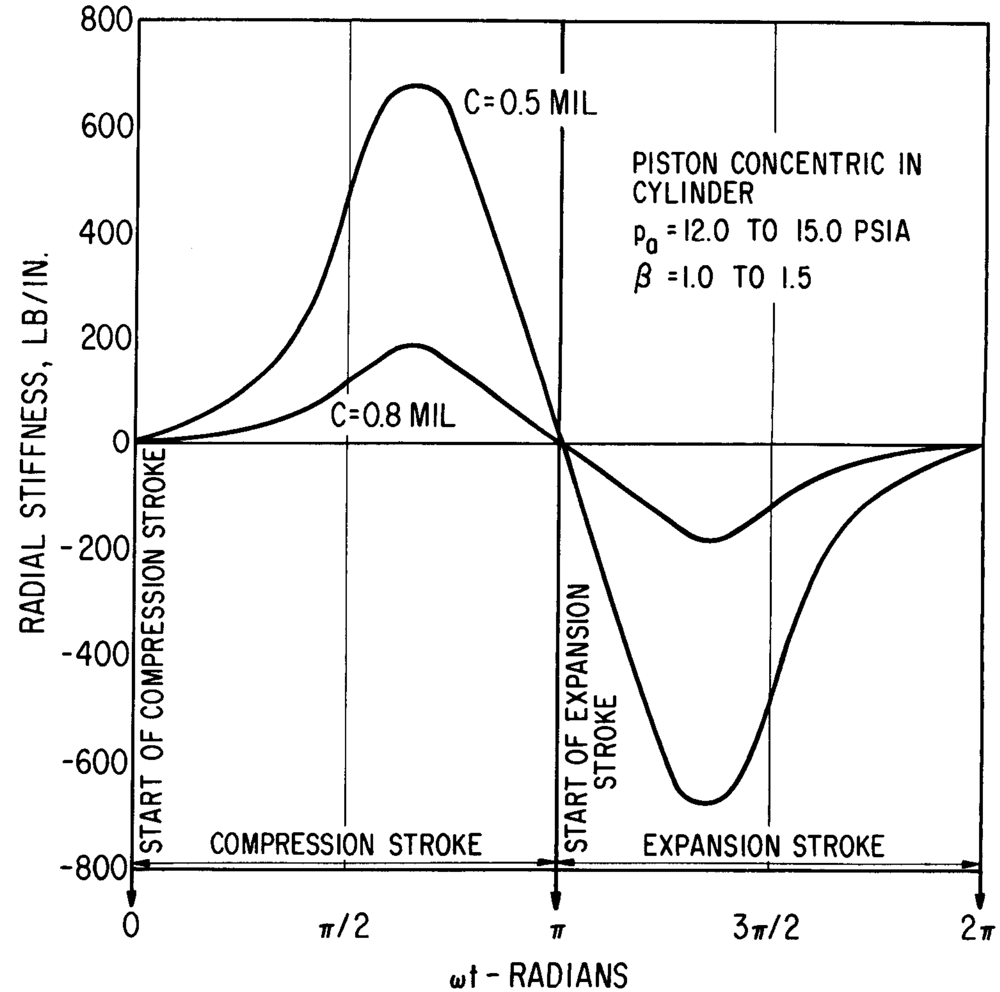



Fig. 8 - Design Map for Resonant Spring




MTI-1747

Fig. 9 - Configuration of the Self-Acting Piston Bearings




MTI-1748

Fig. 10 - Combined Radial Stiffness for the Self-Acting Piston Bearings as a Function of Piston Stroke

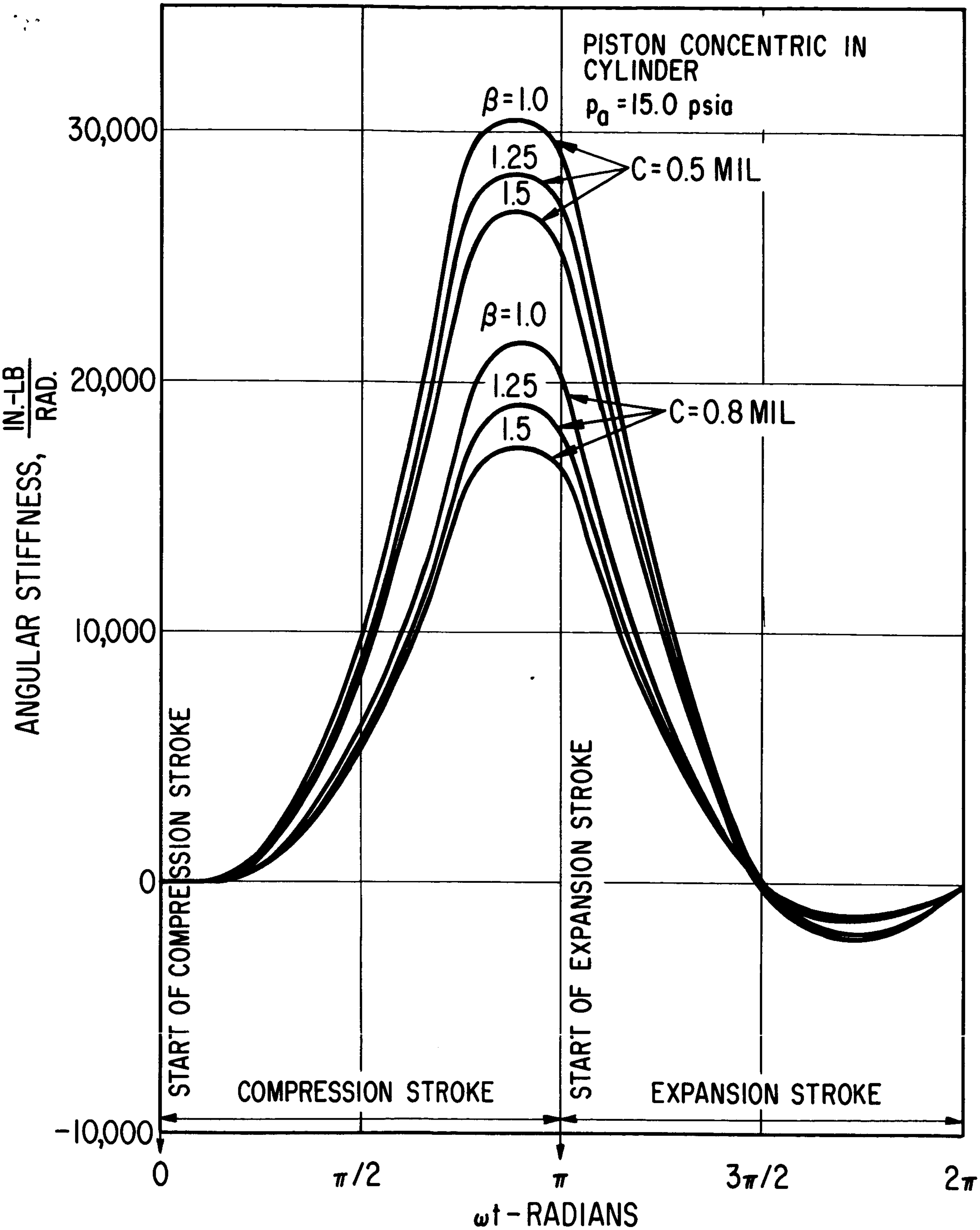
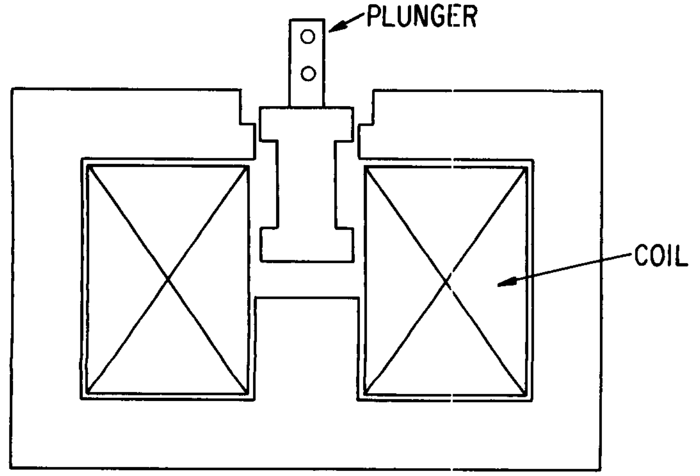
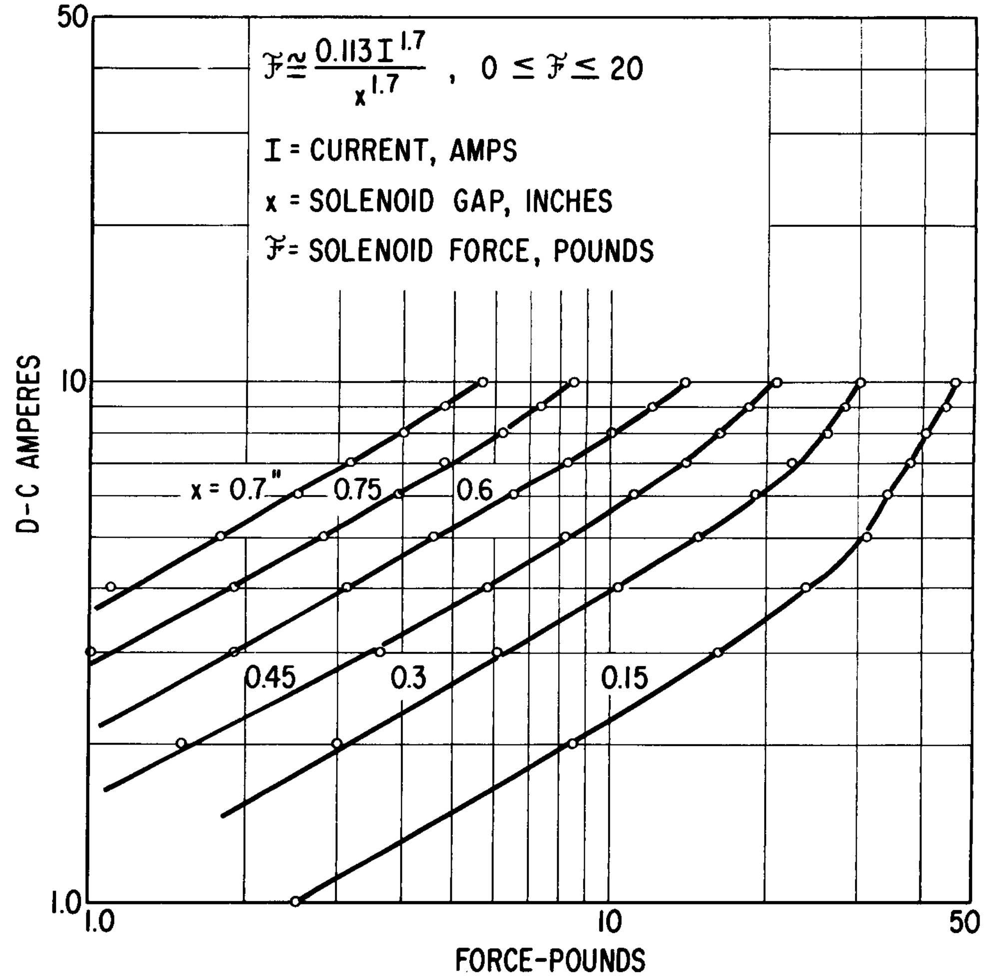



Fig. 11 - Combined Angular Stiffness for the Self-Acting Piston Bearings as a Function of Piston Stroke




MTI-1750

Fig. 12 - Basic Configuration of the Resonant Compressor Drive Solenoid




MTI-1751

Fig. 13 - Measured Force Versus Current Characteristic for the Resenant Compressor

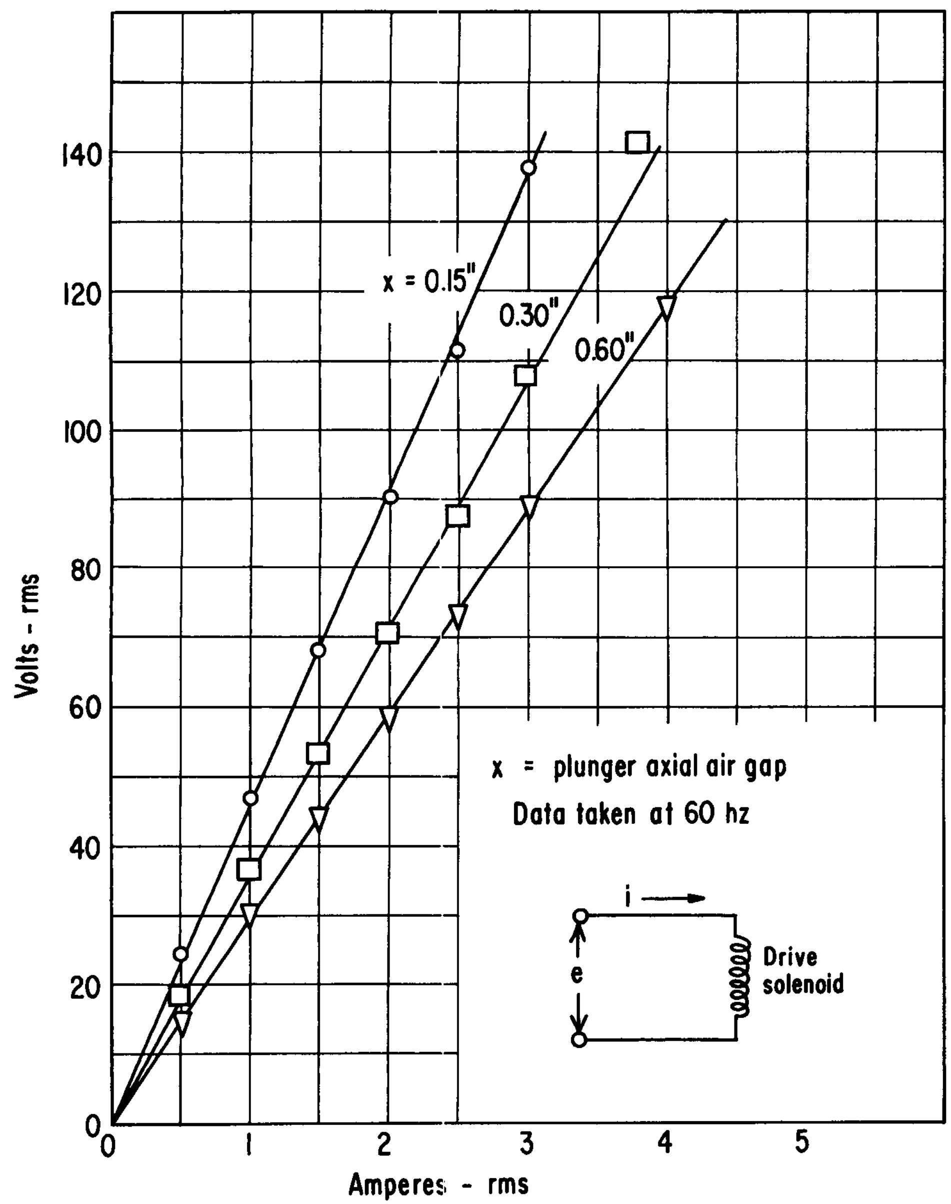



Fig. 14 - Measured 60 HZ RMS Current Versus Voltage for the Resonant Compressor Drive Solenoid

MTI-1752

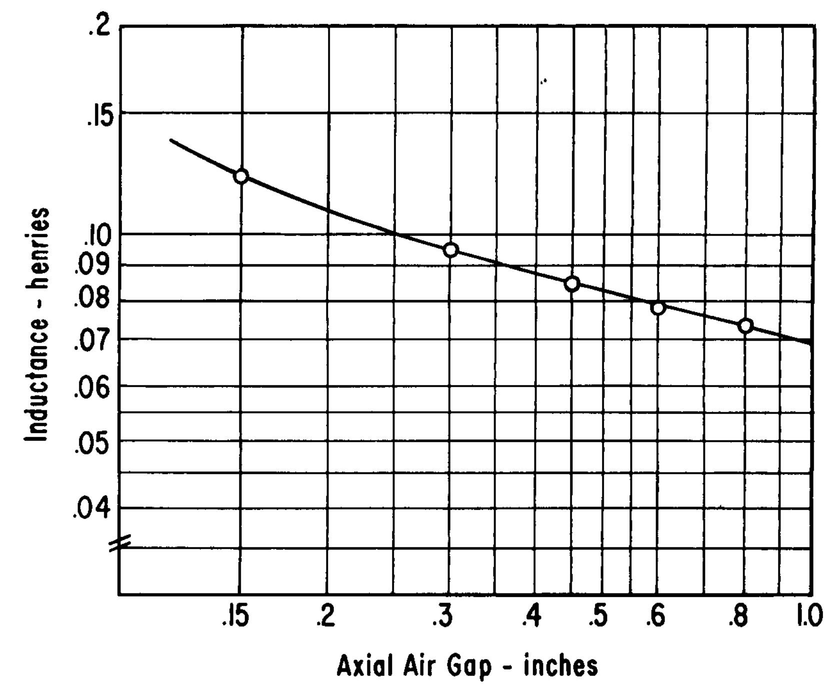



Fig. 15 - Measured Inductance of the Resonant Compressor Drive Solenoid as a Function of Plunger Air Gap

MTI-1753

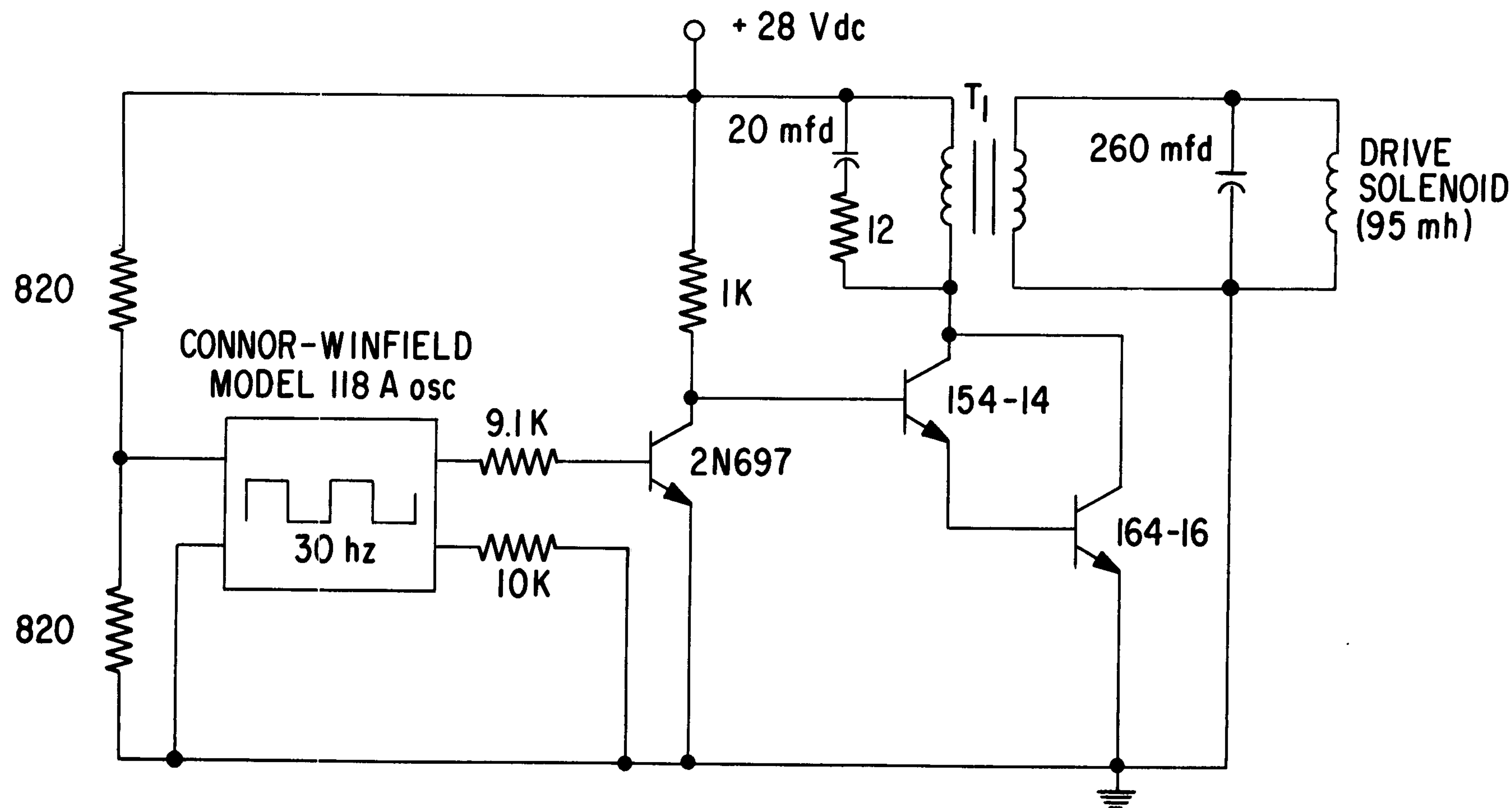



Fig. 16 - Drive Circuit Concept for the Resonant Compressor

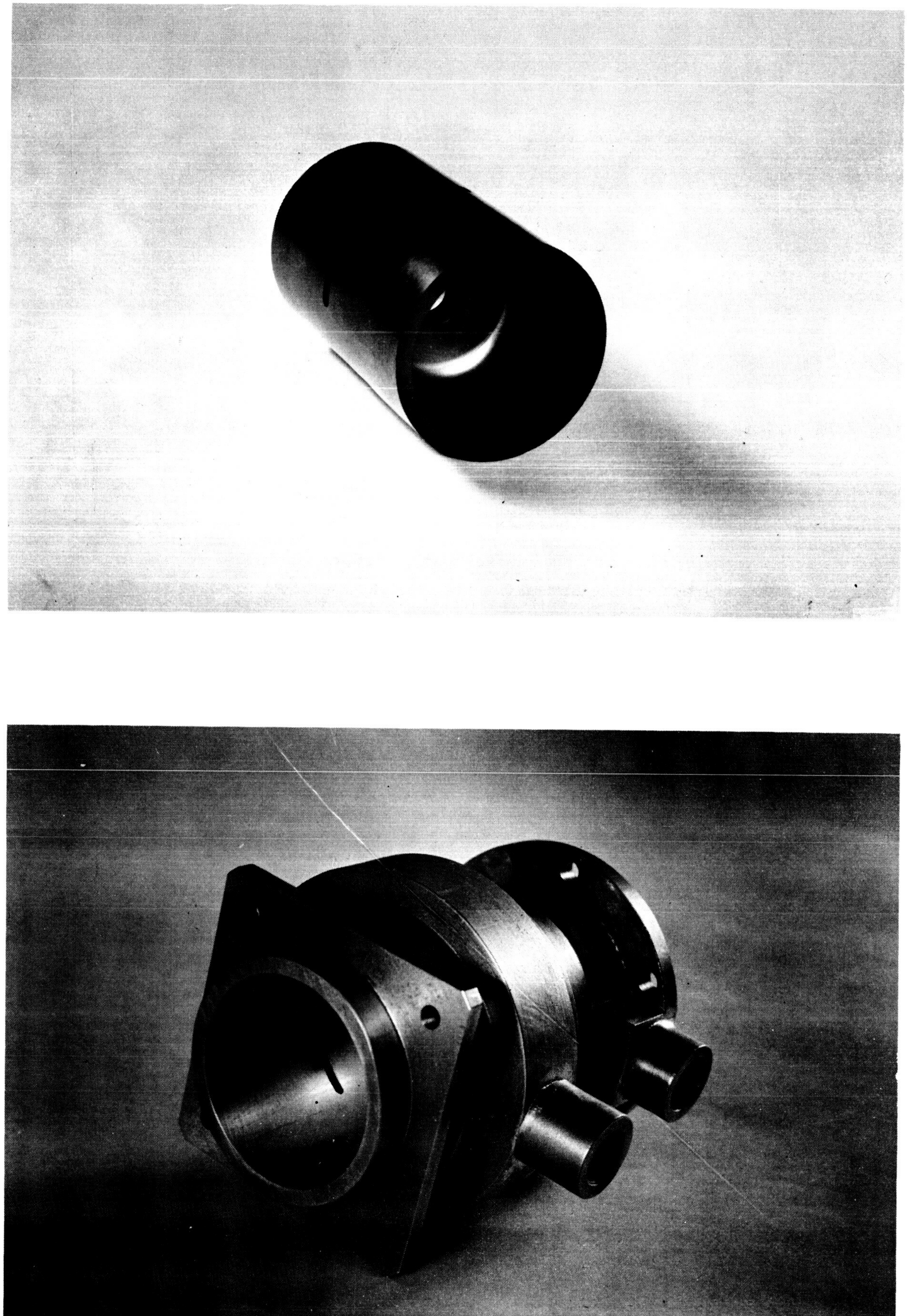


Fig. 17 - Cylinder and Piston for the Resonant Compressor

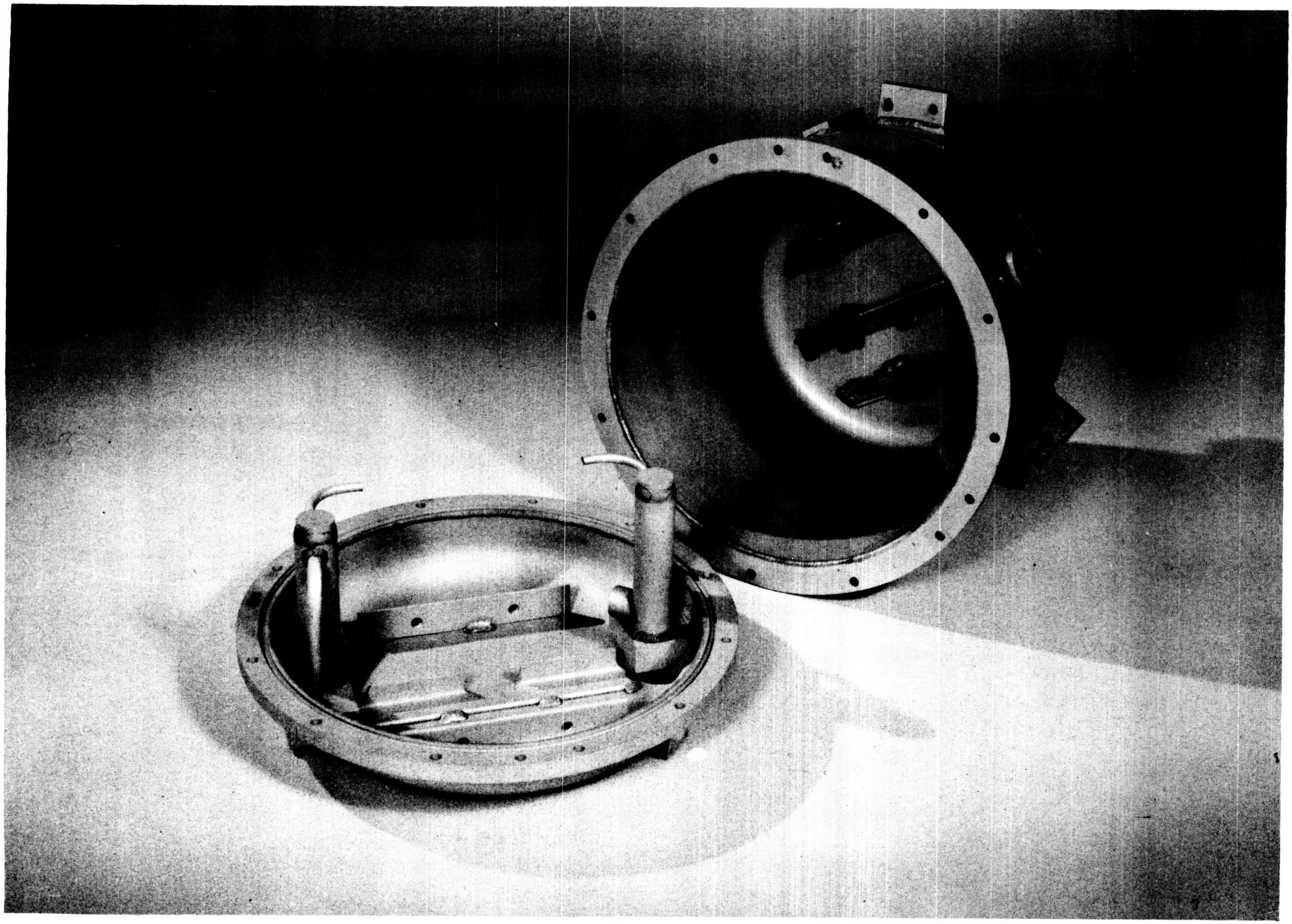


Fig. 18 - Aluminum Container for the Resonant Compressor

MTI-1756

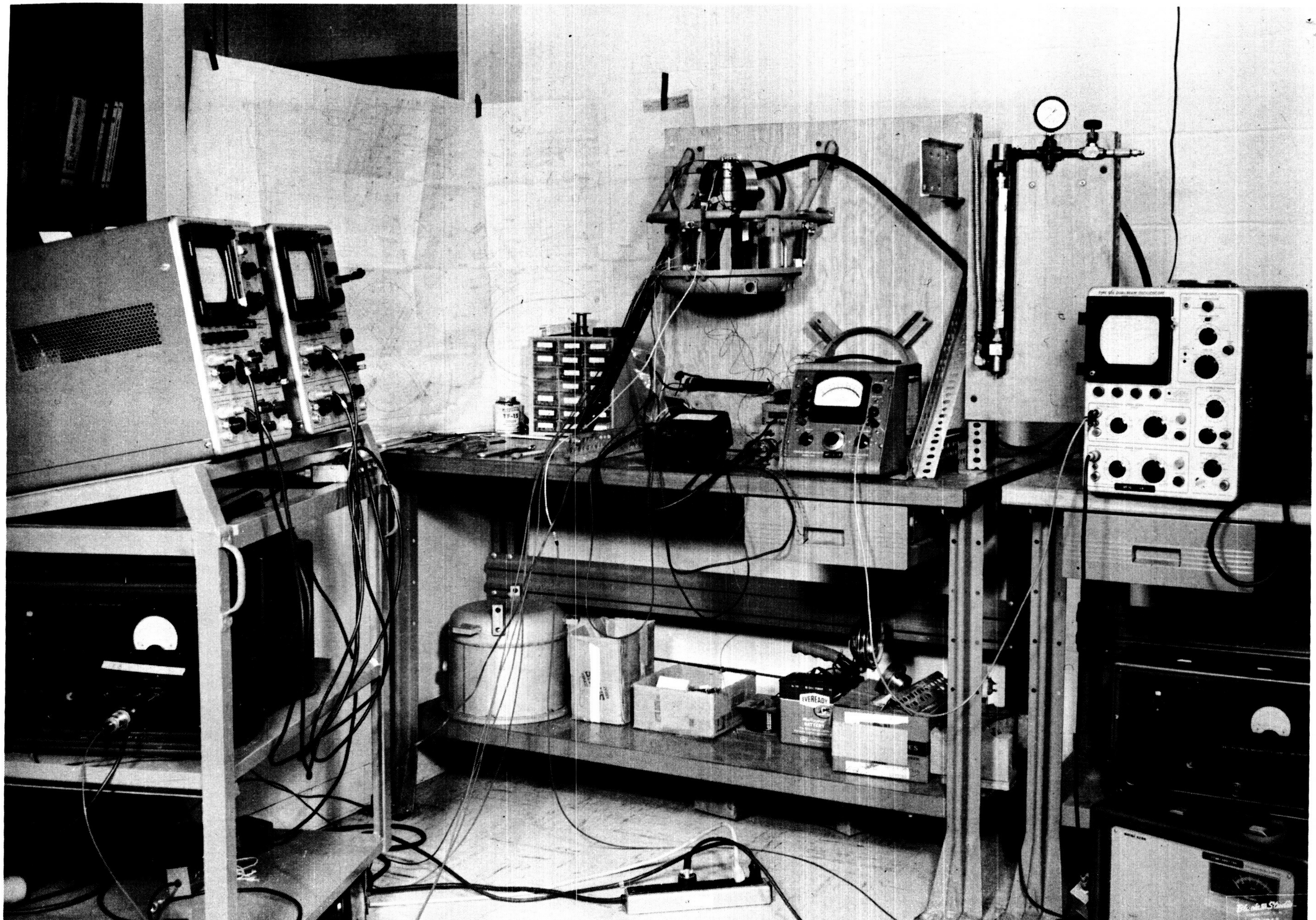



Fig. 19 - View of Resonant Compressor Test Area

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 — RUN NO. 19

MTI PROGRAM NO. RD-272 — DATE 8/28/65

GAS Air

Inlet Pressure 14.7 psia — Pressure Rise 5.0 psi — Flow 0.65 SCFM

Supply Voltage 98.0 60 cps, rms — Total Input Power ______ watts — Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilibrium Position, Suction Direction ______ in.

Piston Amplitude from Equilibrium Position, Discharge Direction ______ in.

Total Stroke ______ in. — Valve Block Temperature 115 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B 07, Rev. 1 — Cylinder Dwg No. 141 D 09, Rev. 2

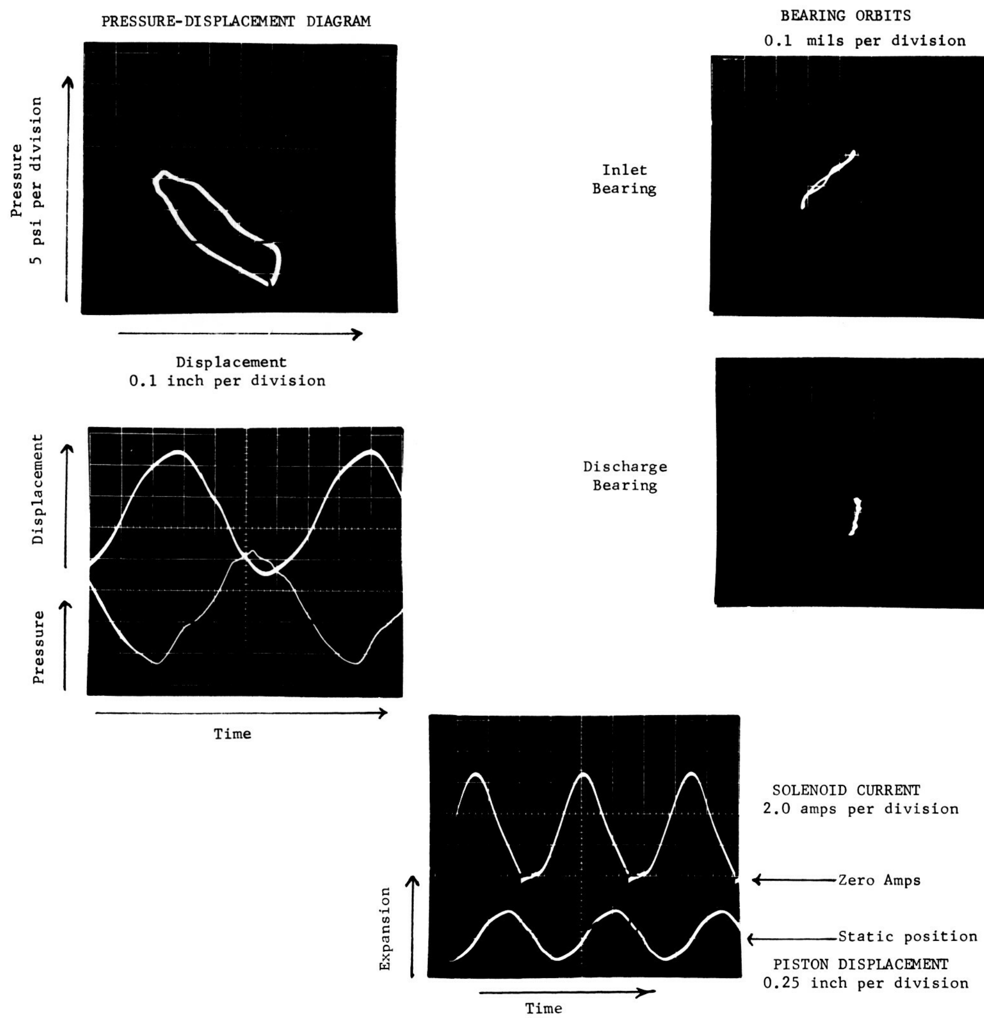



Fig. 20 - Performance Characteristics of First Delivery Compressor at 5 PSI Pressure Rise

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 RUN NO. 21

MTI PROGRAM NO. RD-272 DATE 8/28/65

GAS Air

Inlet Pressure 14.7 psia Pressure Rise 10.0 psi Flow 0.611 SCFM

Supply Voltage 97.5 60 cps, rms Total Input Power ______ watts Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilbrium Position, Suction Direction 0.219 in.

Piston Amplitude from Equilbrium Position, Discharge Direction 0.141 in.

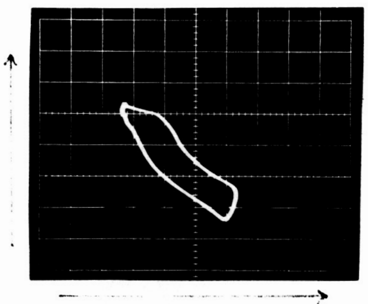
Total Stroke 0.360 in. Valve Block Temperature 124 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B07, Rev. 1 Cylinder Dwg No. 141 D 09, Rev. 2

PRESSURE-DISPLACEMENT DIAGRAM

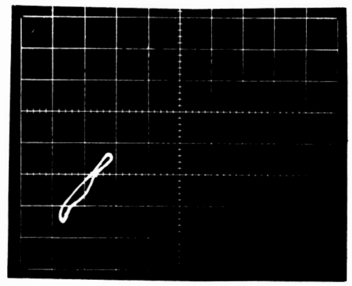
Pressure
5 psi per division



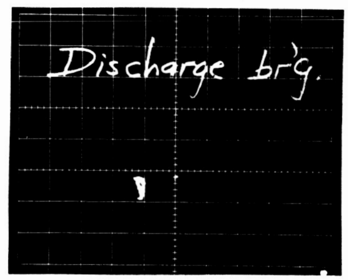
Displacement
0.1 inch per division

BEARING ORBITS
0.1 mils per division

Inlet Bearing

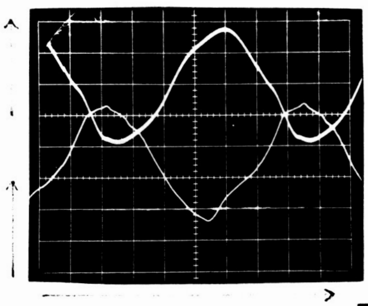


Discharge Bearing




Displacement

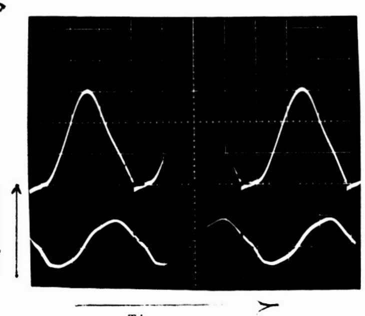
Pressure



Time

SOLENOID CURRENT
2.0 amps per division

Expansion



Zero amps

Static position

PISTON DISPLACEMENT
0.25 inch per division

Time

Fig. 21 - Performance Characteristics of First Delivery Compressor at 10 PSI Pressure Rise

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660

MTI PROGRAM NO. RD-272

RUN NO. 21

DATE 8/28/65

GAS Air

Inlet Pressure 14.7 psia  Pressure Rise 10.0 psi  Flow 0.611 SCFM

Supply Voltage 97.5 60 cps, rms  Total Input Power ______ watts  Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilbrium Position, Suction Direction 0.219 in.

Piston Amplitude from Equilbrium Position, Discharge Direction 0.141 in.

Total Stroke 0.360 in.  Valve Block Temperature 124 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B07, Rev. 1  Cylinder Dwg No. 141 D09, Rev. 2

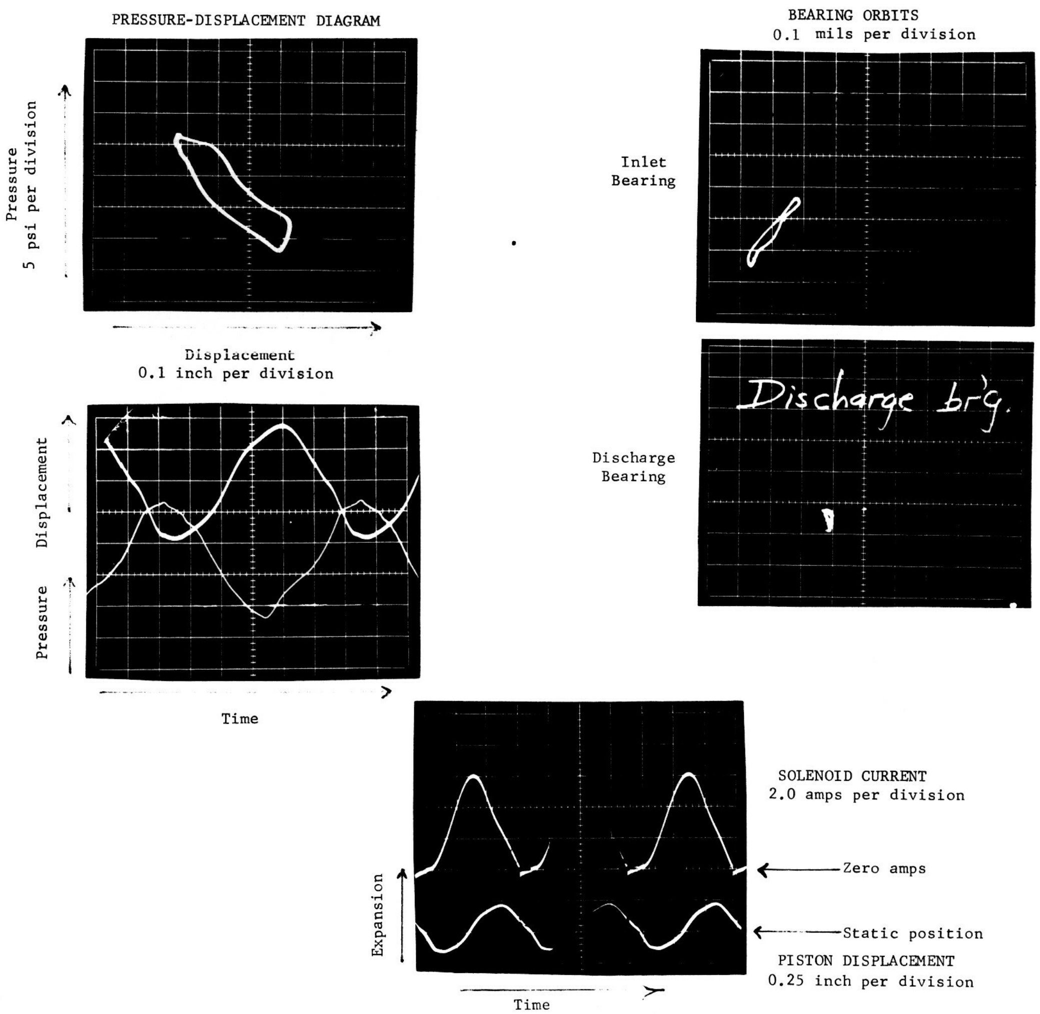



Fig. 21 - Performance Characteristics of First Delivery Compressor at 10 PSI Pressure Rise

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 | RUN NO. 23
MTI PROGRAM NO. RD-272 | DATE 8/28/65
GAS Air

Inlet Pressure 14.7 psia | Pressure Rise 15.0 psi | Flow 0.55 SCFM

Supply Voltage 97.5 60 cps, rms | Total Input Power ______ watts | Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilbrium Position, Suction Direction 0.250 in.

Piston Amplitude from Equilbrium Position, Discharge Direction 0.156 in.

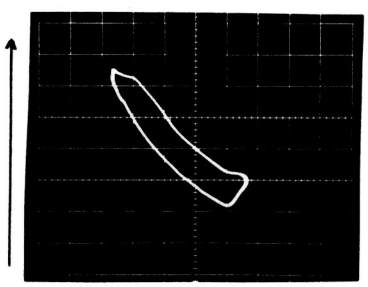
Total Stroke 0.406 in. | Valve Block Temperature 129 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B 07, Rev. 1 | Cylinder Dwg No. 141 D 09, Rev. 2

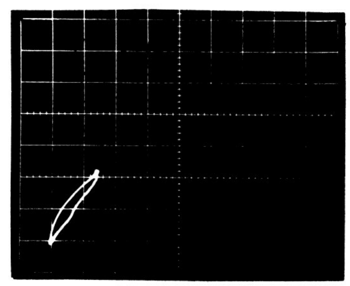
PRESSURE-DISPLACEMENT DIAGRAM

Pressure
5 psi per division



BEARING ORBITS
0.1 mils per division

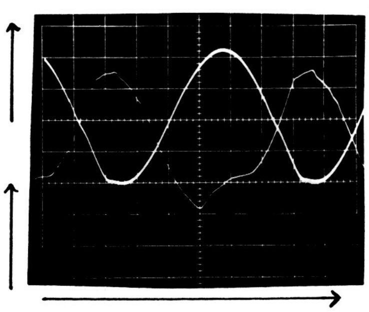
Inlet Bearing



Displacement
0.1 inch per division

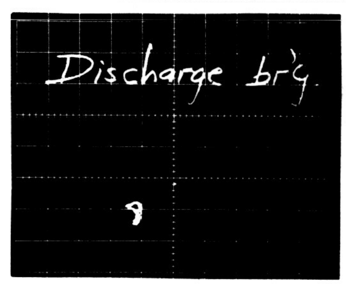
Displacement

Pressure

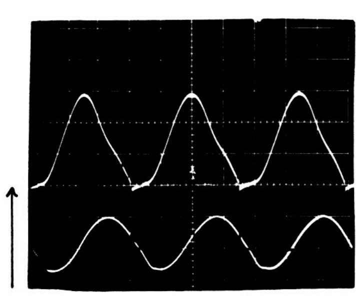


Time

Discharge Bearing




Expansion



SOLENOID CURRENT
2.0 amps per division

Zero amps

Static position

PISTON DISPLACEMENT
0.25 inch per division

Time

Fig. 22 - Performance Characteristics of First Delivery Compressor at 15 PSI Pressure Rise

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660

MTI PROGRAM NO. RD-272

RUN NO. 24

DATE 8/28/65

GAS Air

Inlet Pressure 14.7 psia    Pressure Rise 17.5 psi    Flow 0.522 SCFM

Supply Voltage 98.0 60 cps, rms    Total Input Power ______ watts    Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilbrium Position, Suction Direction 0.281 in.

Piston Amplitude from Equilbrium Position, Discharge Direction 0.172 in.

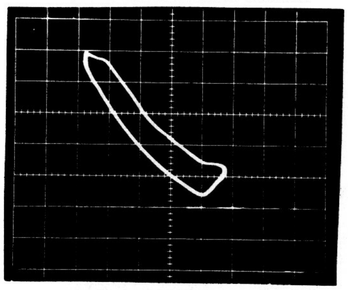
Total Stroke 0.453 in.    Valve Block Temperature 131 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B 07, Rev. 1    Cylinder Dwg No. 141 D 09, Rev. 2

PRESSURE-DISPLACEMENT DIAGRAM

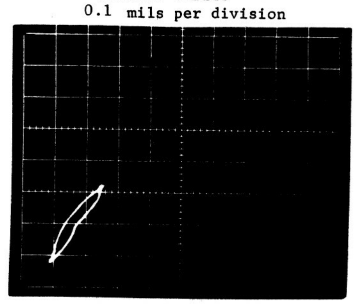
Pressure
5 psi per division



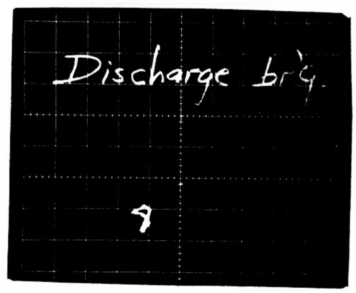
Displacement
0.1 inch per division

BEARING ORBITS
0.1 mils per division

Inlet Bearing

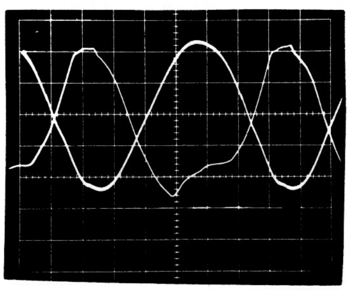


Discharge Bearing




Displacement

Pressure



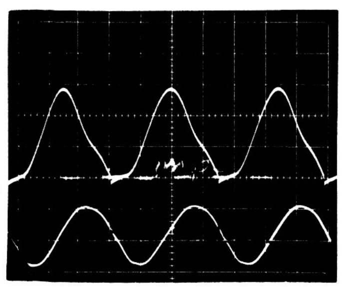
Time

SOLENOID CURRENT
2.0 amps per division

Zero amps

Static position

Expansion



PISTON DISPLACEMENT
0.25 inch per division

Time

Fig. 23 - Performance Characteristics of First Delivery Compressor at 17.5 PSI Pressure Rise

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660

MTI PROGRAM NO. RD-272

RUN NO. 25

DATE 8/28/65

GAS Air

Inlet Pressure 14.7 psia    Pressure Rise 20.0 psi    Flow 0.529 SCFM

Supply Voltage 98.0 60 cps, rms    Total Input Power ______ watts    Overall Eff. ______ %

______ d-c

Piston Amplitude from Equilbrium Position, Suction Direction 0.312 in.

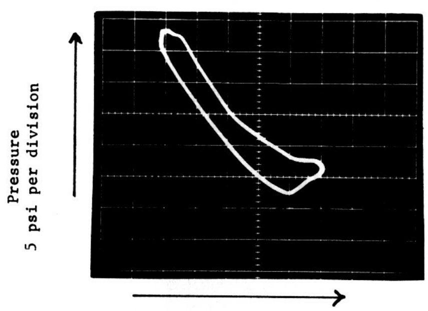
Piston Amplitude from Equilbrium Position, Discharge Direction 0.187 in.

Total Stroke 0.499 in.    Valve Block Temperature 135 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141 B 07, Rev. 1    Cylinder Dwg No. 141 D 09, Rev. 2

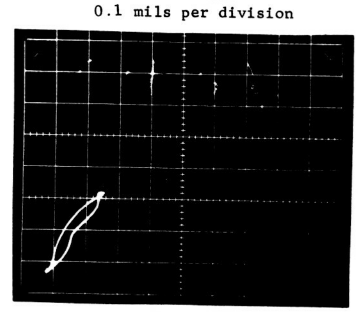
PRESSURE-DISPLACEMENT DIAGRAM



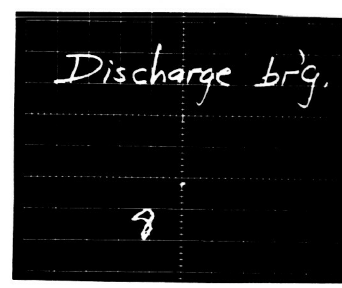

Displacement
0.1 inch per division

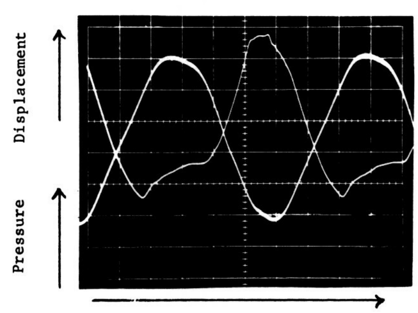
BEARING ORBITS
0.1 mils per division

Inlet
Bearing

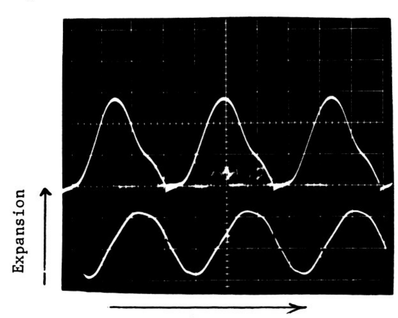


Discharge
Bearing




Time




SOLENOID CURRENT
2.0 amps per division

Zero amps

Static position

PISTON DISPLACEMENT
0.25 inch per division

Time

Fig. 24 - Performance Characteristics of First Delivery Compressor at 20 PSI Pressure Rise

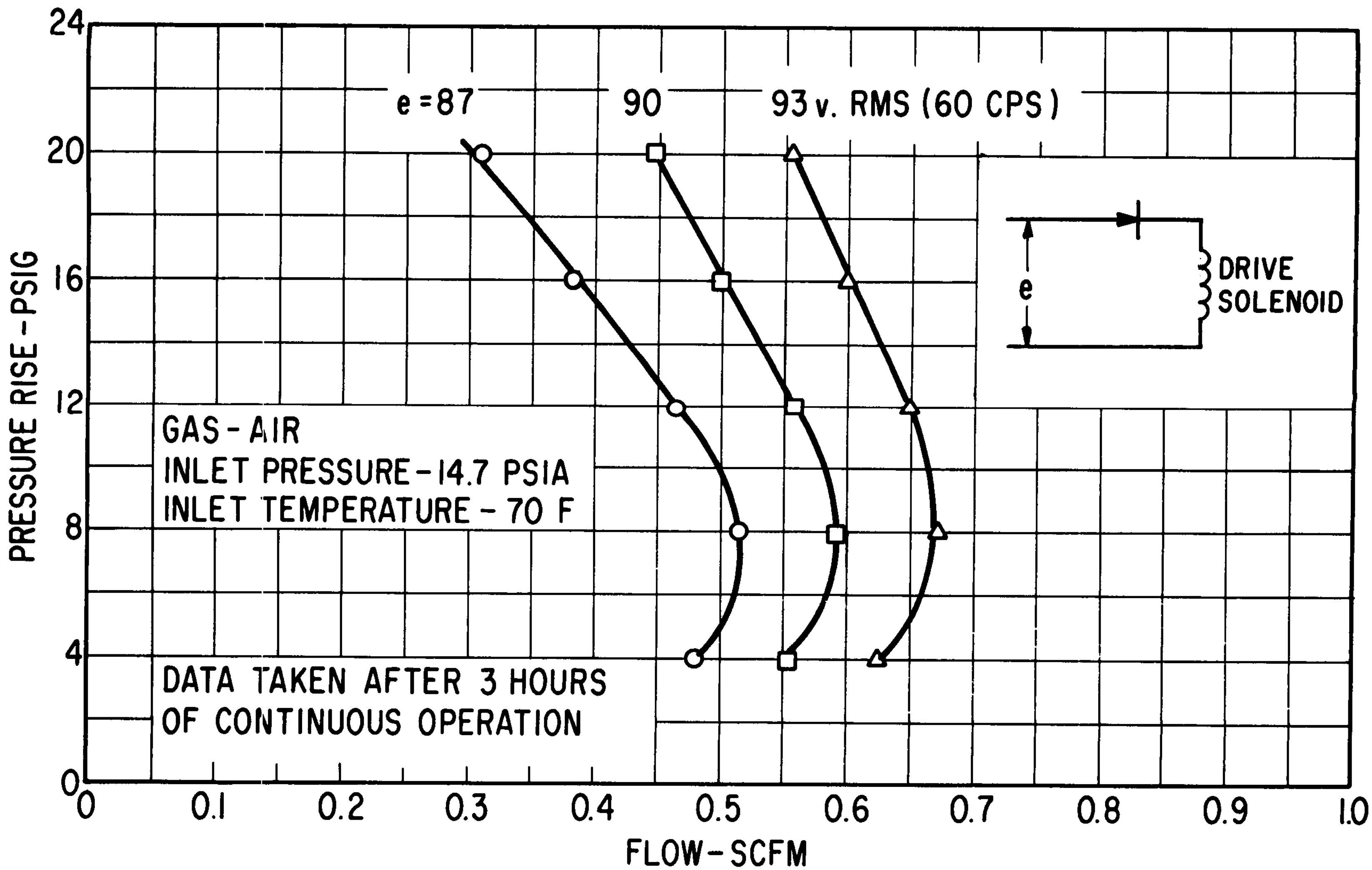



Fig. 25 - Flow Versus Pressure Rise as a Function of Solenoid Voltage for the First Delivery Compressor

6-day run, spare compressor (9th hour of test)

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 — RUN NO. 51

MTI PROGRAM NO. RD-272 — DATE 4-14-66

Room Temperature 85 F — GAS Air

Inlet Pressure 14.7 psia — Pressure Rise 17.0 psi — Flow 0.483 SCFM

Supply Voltage 85.5 60 cps, rms — Total Input Power ______ watts — Overall Eff. ______ %

______ d-c — Solenoid Coil-surface Temperature 108 F

Piston Amplitude from Equilbrium Position, Suction Direction ______ in.

Piston Amplitude from Equilbrium Position, Discharge Direction ______ in.

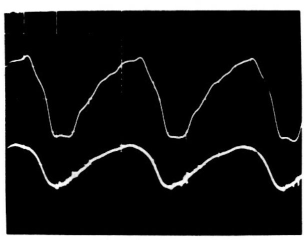
Total Stroke ______ in. — Valve Block Temperature 143 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

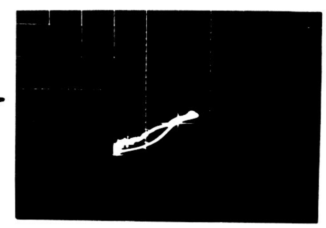
Piston Dwg No. 141B07, Rev. 1 — Cylinder Dwg No. 141D09, Rev. 2

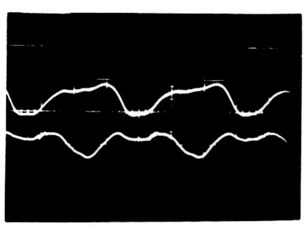
BEARING X-Y TIME
BASE TRACES
0.1 mils per vertical division

BEARING ORBITS
0.1 mils per division

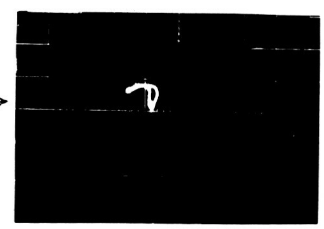


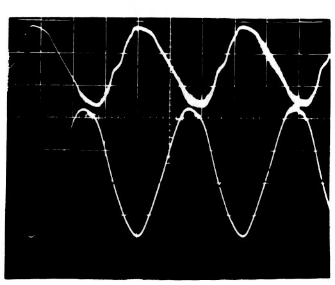
← Inlet Bearing →





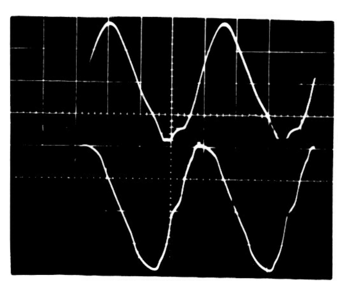
← Discharge Bearing →





Solenoid voltage

Piston displacement



Solenoid current

← zero amps

Solenoid voltage

→ Time

→ Time

Fig. 26 - Performance Characteristics of Spare Compressor at 9th Hour of 153 Hour Continuous Run

6-day run, spare compressor (120th hour of test)

## RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 RUN NO. 61

MTI PROGRAM NO. RD-272 DATE 4-19-66

GAS Air

Room Temperature 75 F

Inlet Pressure 14.7 psia Pressure Rise 17.0 psi Flow 0.541 SCFM

Supply Voltage 86.9 60 cps, rms Total Input Power ______ watts Overall Eff. ______ %

______ d-c Solenoid Coil-surface Temperature 107 F

Piston Amplitude from Equilbrium Position, Suction Direction ________ in.

Piston Amplitude from Equilbrium Position, Discharge Direction _______ in.

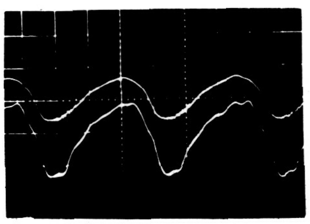
Total Stroke _______ in. Valve Block Temperature 139 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

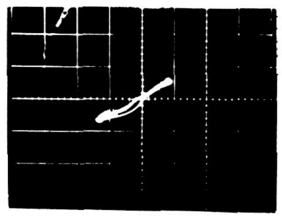
Piston Dwg No. 141B07, Rev. 1 Cylinder Dwg No. 141D09, Rev. 2

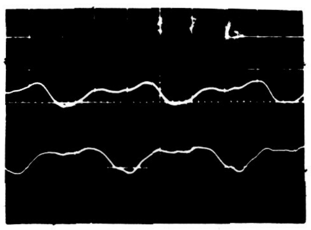
BEARING X-Y TIME
BASE TRACES
0.1 mils per vertical division

BEARING ORBITS
0.1 mils per division

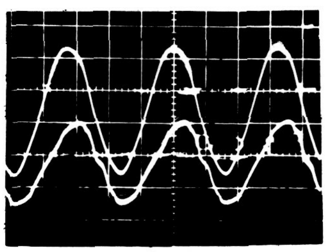


← Inlet Bearing →





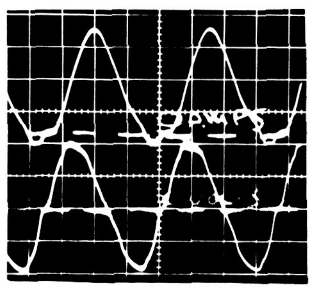
← Discharge Bearing →



Piston displacement

Solenoid voltage

Time →



Solenoid current

← zero amps

Solenoid voltage

Time →

Fig. 27 - Performance Characteristics of Spare Compressor at 120th Hour of 153 Hour Continuous Run

6-day run, spare compressor (150th hour of test)

# RESONANT COMPRESSOR PERFORMANCE DATA

NASA CONTRACT NO. NAS 8-11660 RUN NO. 63

MTI PROGRAM NO. RD-272 DATE 4-20-66

Room Temperature 78 F GAS Air

Inlet Pressure 14.7 psia Pressure Rise 16.9 psi Flow 0.543 SCFM

Supply Voltage 86.9 60 cps, rms Total Input Power ______ watts Overall Eff. ______ %

______ d-c Solenoid Coil-surface Temperature 107 F

Piston Amplitude from Equilbrium Position, Suction Direction ________ in.

Piston Amplitude from Equilbrium Position, Discharge Direction _______ in.

Total Stroke _______ in. Valve Block Temperature 139 °F

Distance from End of Cylinder to End of Piston in Equilibrium Position 0.328 in.

Piston Dwg No. 141B07, Rev. 1 Cylinder Dwg No. 141D09, Rev. 2

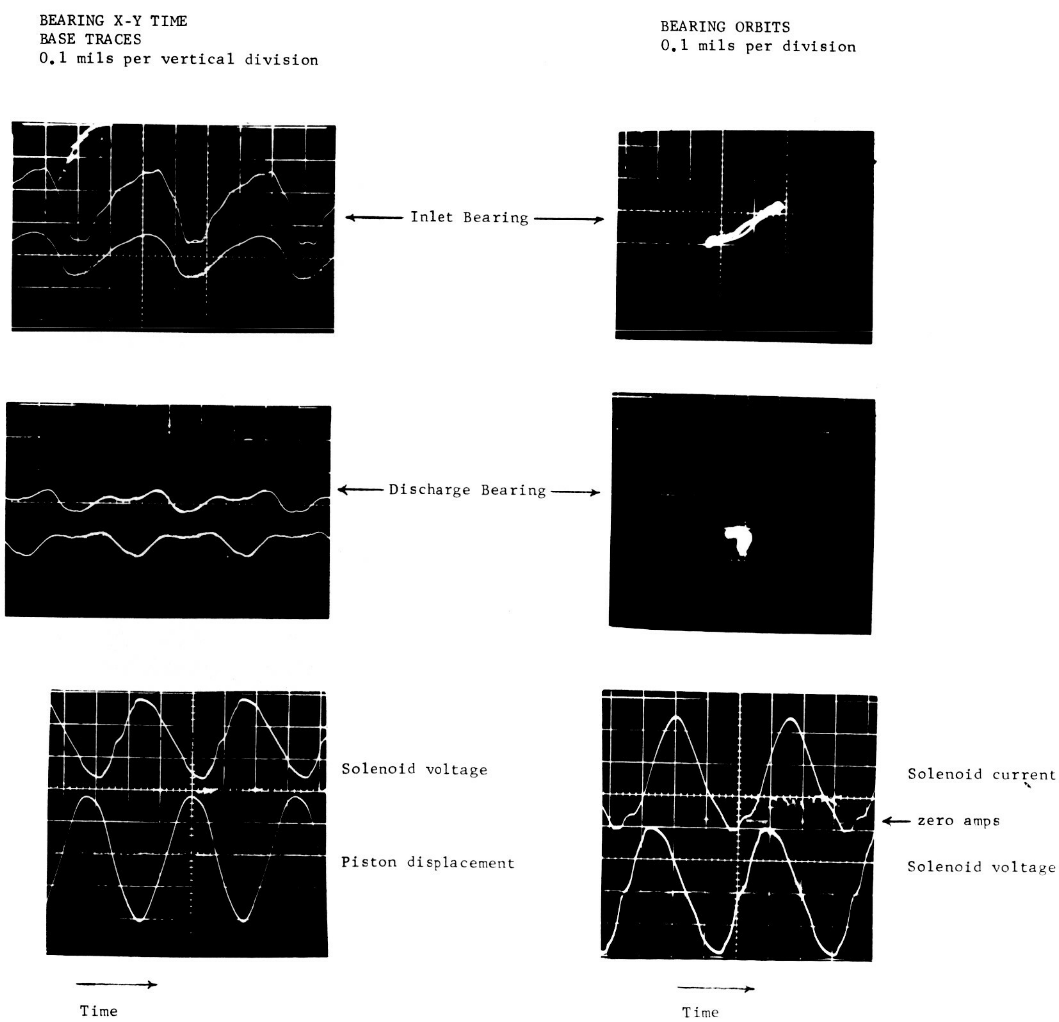
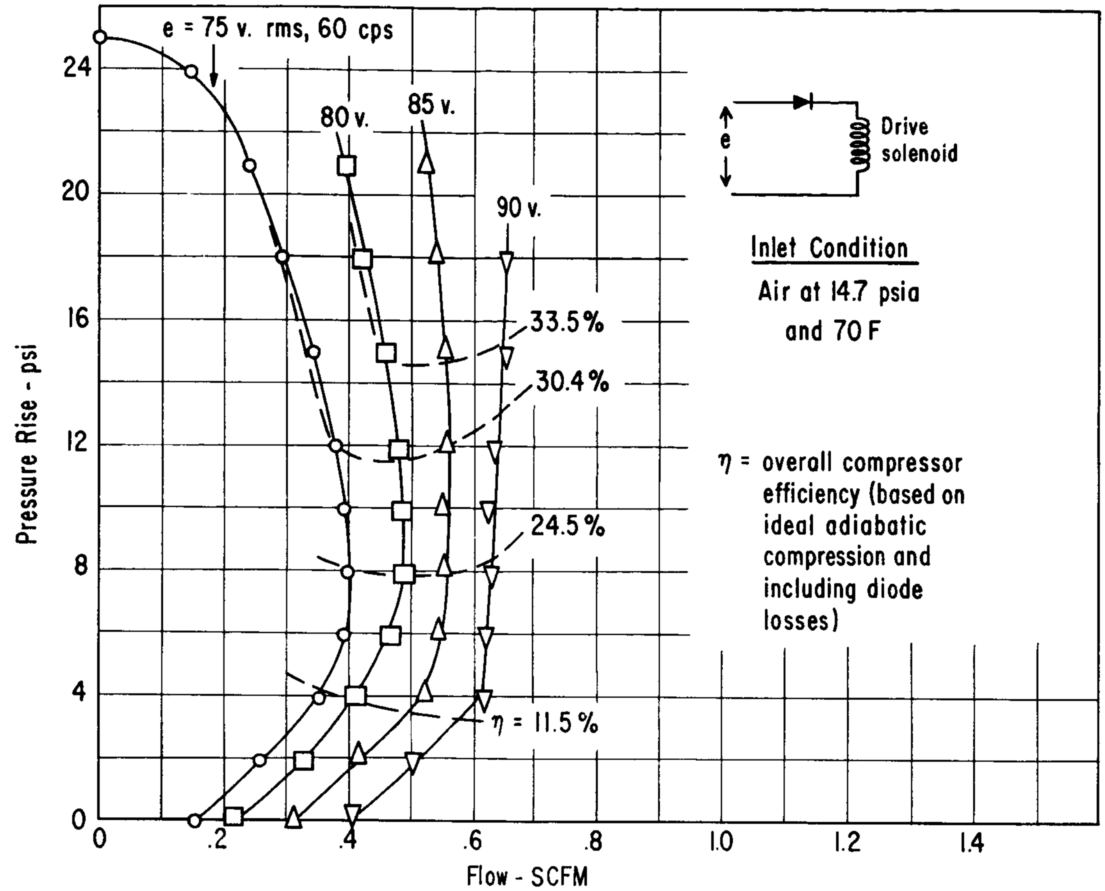
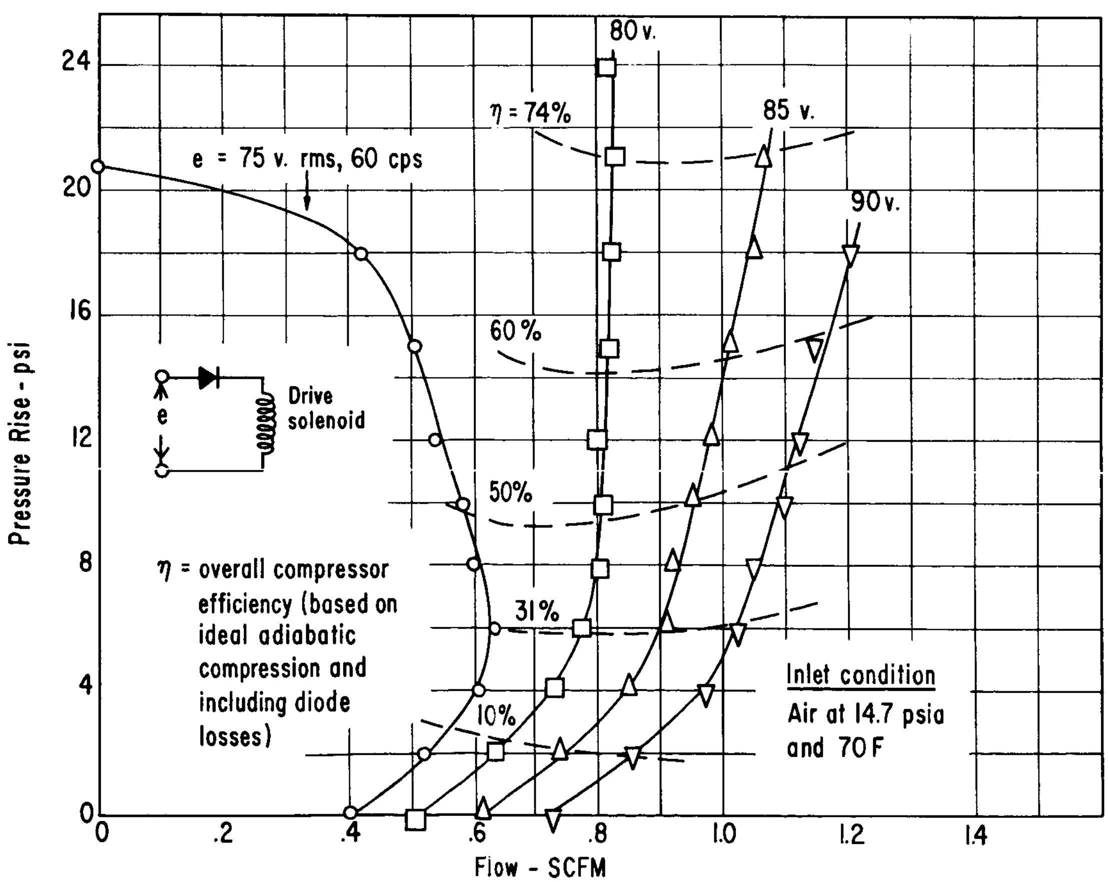



Fig. 28 - Performance Characteristics of Spare Compressor at 150th Hour of 153 Hour Continuous Run



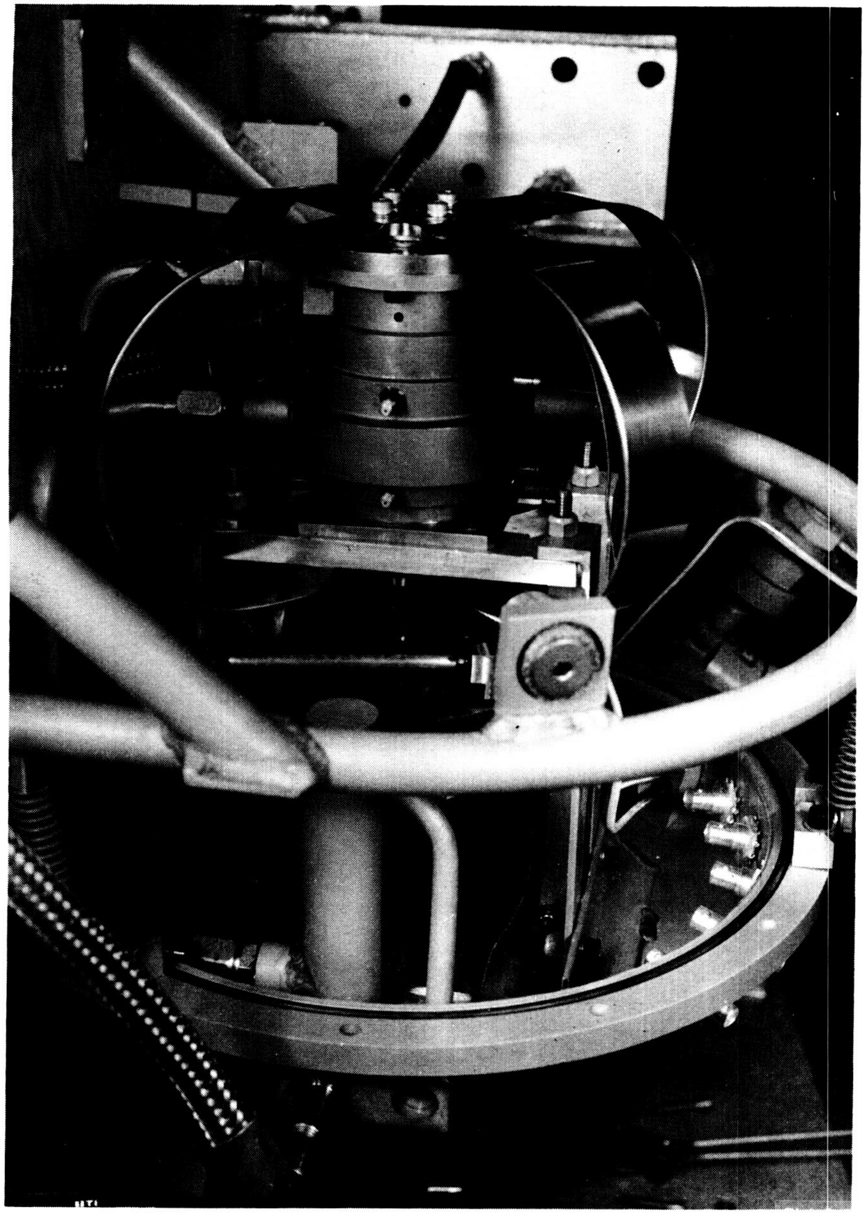

MTI-1767

Fig. 29 - Flow Versus Pressure Rise as a Function of Solenoid Voltage for the Spare Compressor



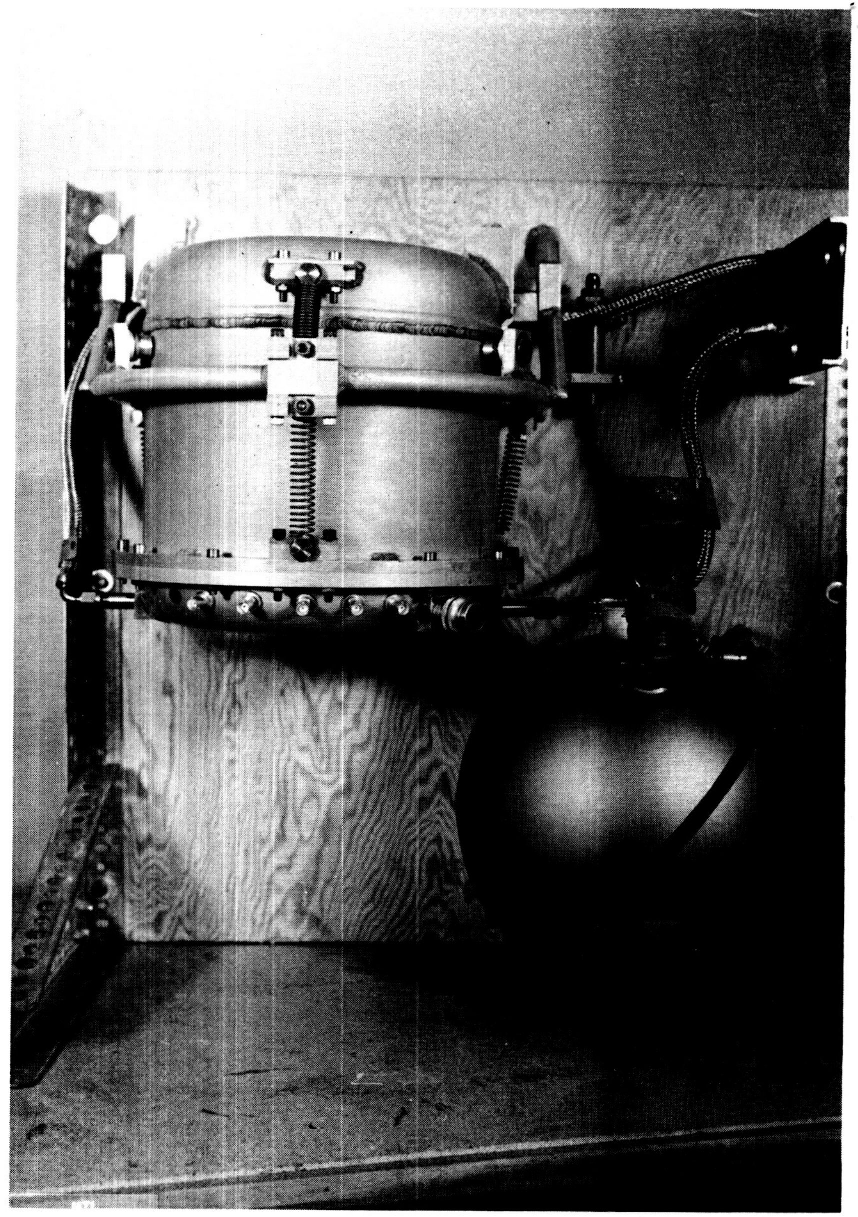

MTI-1768

Fig. 30 - Flow Versus Pressure Rise as a Function of Solenoid Voltage for the Spare Compressor, with Inlet and Discharge Gas Coolers By-Passed



MTI-1769

Fig. 31 - View of Spare Compressor With Top Section of Container Removed



MTI-1770

Fig. 32 - View of Spare Compressor and Gas Make-Up Subsystem

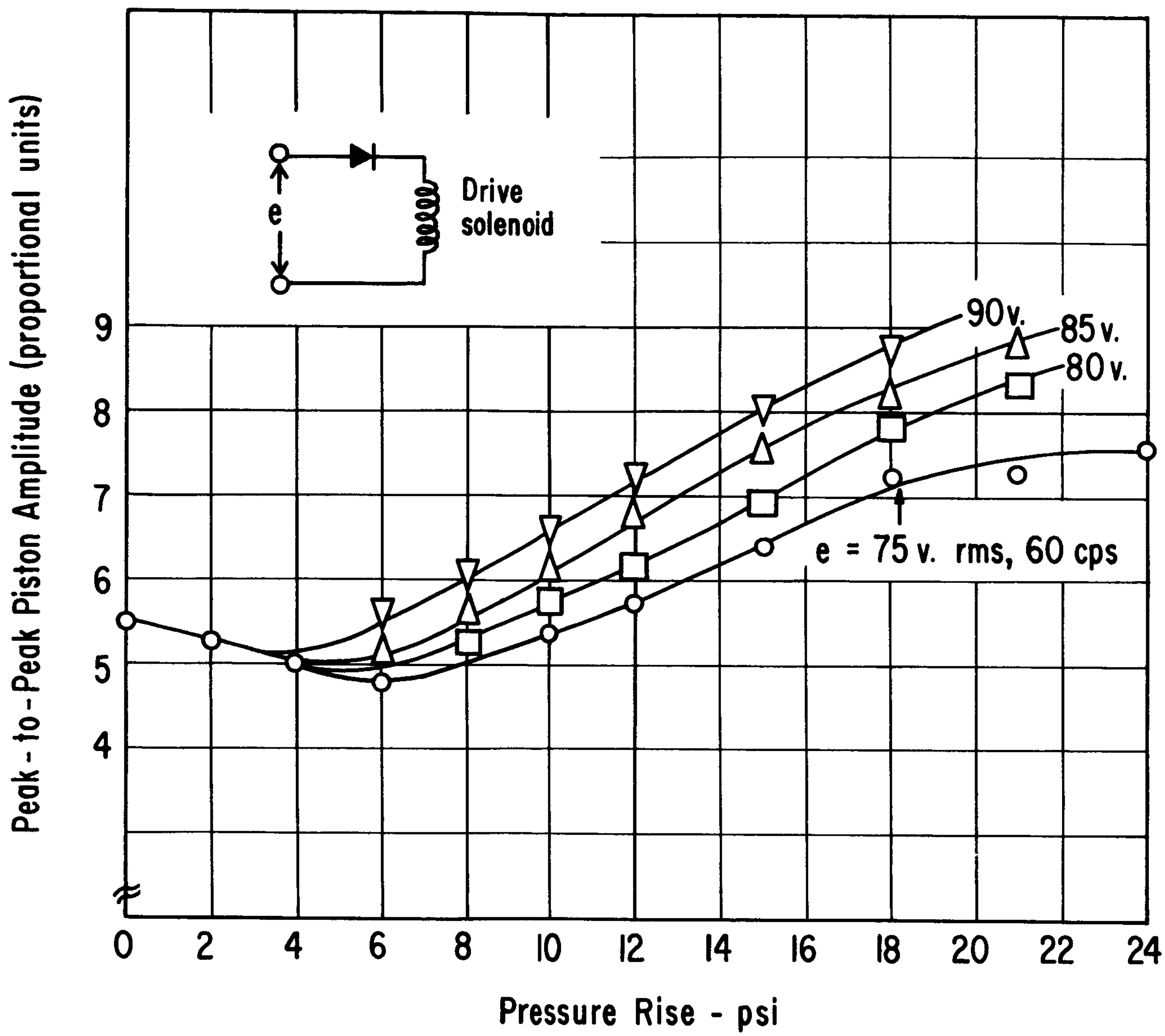



Fig. 33 - Change in Piston Stroke As a Function of Compressor Pressure Rise and Drive Solenoid Voltage

# APPENDIX A

CONTRACT NAS8-11660
MECHANICAL TECHNOLOGY INC.

EXHIBIT "A"

SCOPE OF WORK

PART I - STATEMENT OF WORK

SATURN V research and development - gas supply system for hermetically sealed platforms.

1. Design a gas supply system including compressor to the specifications outlined below.

2. Manufacture and deliver one complete prototype system.

3. Manufacture and deliver one compressor complete with fittings suitable for use as a spare compressor for the system.

4. System Operation:

a. Figure 1 is included to illustrate a possible component arrangement in order to clarify the type of system desired. It is not intended to fix the design. This figure will be used as a reference to briefly describe the system operation.

b. Gas of the desired specifications is to be supplied to the stabilized platform. The gas, after flowing through the gas bearing components, will be ejected into the void inside the platform cover; and the system loop is closed by returning this ejected gas to the input of the compressor.

c. If the pressure inside the platform cover drops below a specified limit, replenishment of gas from the gas reservoir into the platform cover is activated and is deactivated when a specified upper limit of pressure is reached.

5. Physical Specification (System):

Size - 28" square panel (approx. - no larger)
Weight - minimum to achieve other specifications
Mounting - optional
Ambient temperature operating range - 0°C + 60°C
Ambient pressure - $10^{-9}$ Torr (space vacuum)

Scope of Work (Cont'd)

6. Compressor Specifications:

   $P_1$ = input pressure range 12 to 15 psia
   Gas flow = 14 to 15 liters/min
   Output to system = capable of monitoring 15 psig = ½ at 0.50 scfm, as measured across platform

7. Gas Supply Specifications:

   Gas reservoir - cut in at 12 = ½ psia absolute pressure
   cut out at 15 = ½ psia absolute pressure

   A ½ cu. ft. volume titanium sphere shall be used for the gas reservoir. Sources for these spheres include:

   (a) Menasco Mfg. Co., 805 S. Fernando Blvd., Burbank, California

   (b) Airite Products, Div. of Electrada Corp., 3518 East Olympic Blvd., Los Angeles, California

   Output temperature of gas: 15°C ± 5°C
   Dew point: less than minus 54°C
   Dust particles - no greater than 8 microns
   Condensible hydrocarbon content - less than 1 part/million by weight

8. System Vibration Requirements:

   Linear acceleration - 10g
   Log sweep - 15 minutes
   0-30 cps at 0.2 inch double amplitude
   30-2,000 cps at 5g
   Shock - 50g for 11 milliseconds

9. Electrical Power:

   Voltage - either 26 volts, 400 cycle ac or 28 V dc
   Power - 150 watts maximum

10. Operating Life - 10,000 hours.

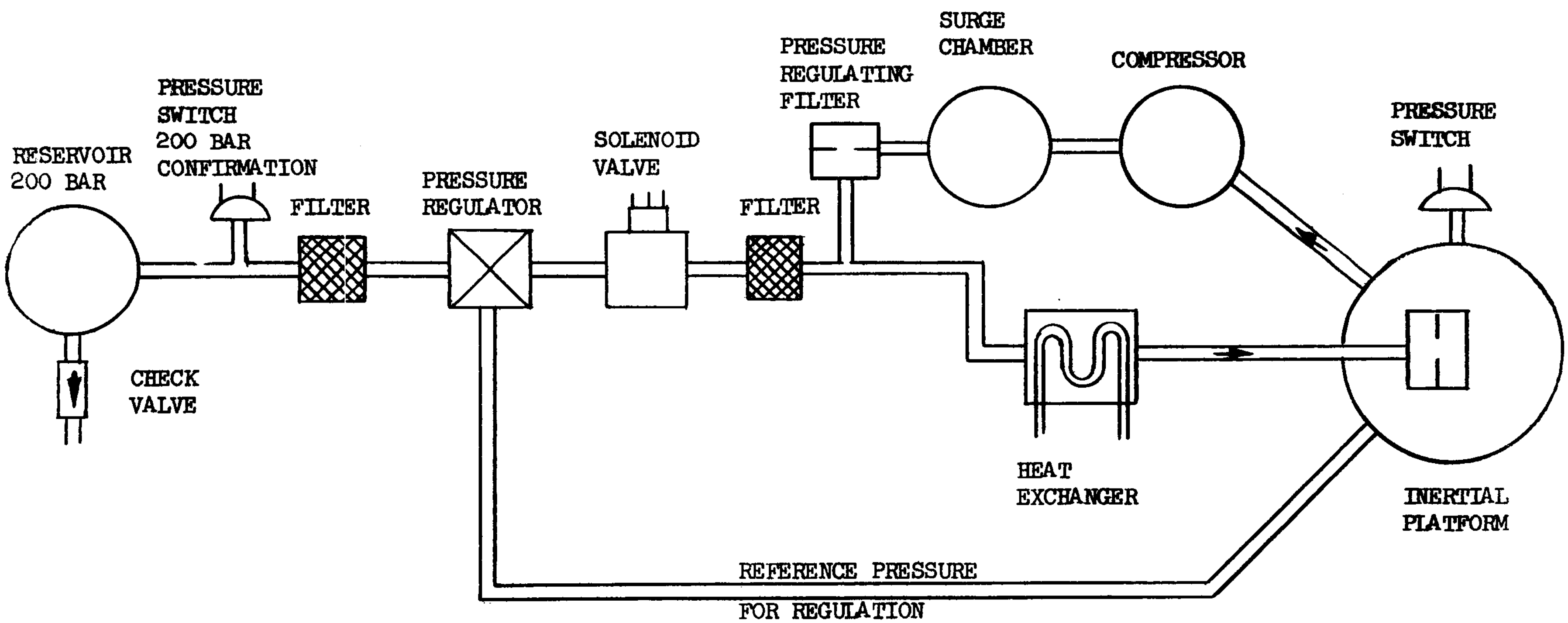



FIGURE I RECIRCULATING GAS SUPPLY SYSTEM

# APPENDIX B

## COMPONENT SPECIFICATIONS FOR GAS SUPPLY SYSTEM AS SHOWN IN FIGURE 1
(Reference Drawing SK-A-1889)

### General Conditions For Each Component

NUMBER REQUIRED PER SYSTEM - 1 each

SIZE - Minimum consistent with reliable design

WEIGHT - Minimum consistent with reliable design

AMBIENT TEMPERATURE RANGE - $0^{o}C$ to $+60^{o}C$

AMBIENT PRESSURE - Sea Level to $10^{-9}$ Torr (Space Vacuum)

VIBRATION REQUIREMENTS - Linear Acceleration - 10g
Log Sweep - 15 minutes
0 - 30 cps at 0.2 inch double amplitude
30 - 2000 cps at 5g
Shock - 50g for 11 milliseconds

OPERATING LIFE - 10,000 hours

EQUIPMENT FOR SATURN V DEVELOPMENT

CLEANING SPECIFICATION - MSFC-1041-9906A

LUBRICATION SHOULD NOT BE NECESSARY FOR OPERATION.

### Item D - Low Pressure Filter

TEMPERATURE - $15^{o}C \pm 5^{o}C$

INPUT PRESSURE - 31 PSIA

CLEAN PRESSURE DROP - 0.1 PSI maximum at flow of
0.5 SCFM nitrogen gas

REMOVAL RATING - 98% 1.5 microns
100% 15.0 microns

RELIEF VALVE - None

EXTERNAL LEAKAGE - Zero

PROOF PRESSURE - 60 PSIA

BURST PRESSURE - 75 PSIA

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

MATERIAL - Anodized aluminum or stainless steel.

Item E - Differential Pressure Regulator

INLET TEMPERATURE - $15^{o}C \pm 5^{o}C$

FLOW - 0.5 SCFM nitrogen gas

Dew point: Less than $-54^{o}C$

PRESSURE SCHEDULE:

INPUT PRESSURE RANGE - 28 to 31 PSIA

OUTPUT PRESSURE RANGE - Capable of providing a pressure increase over the compressor inlet (compressor inlet range of 12 PSIA to 15 PSIA) of 15 $\pm$ 1/2 PSIG

RESPONSE TIME -

VALVE TO FAIL OPEN

NO CONSTANT REFERENCE PRESSURE. THEREFORE VALVE IS TO HOLD CONSTANT $\Delta$P OF 15 $\pm$ 1/2 PSIG.

LIFE - 10,000 hours

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

DOME LOADING PORT - AND-10050-4

PROOF PRESSURE - 60 PSIA

BURSH PRESSURE - 75 PSIA

EXTERNAL LEAKAGE - Zero

INTERNAL LEAKAGE - Zero

MATERIAL - Anodized aluminum or stainless steel.

Item F - Restrictor

INLET PRESSURE - 27 to 30 PSIA

OUTLET PRESSURE - 12 to 15 PSIA

FLOW - 0.005 SCFM nitrogen gas

GAS TEMPERATURE - $0^{o}C$ to $15^{o}C \pm 5^{o}C$

RESTRICTOR TO ACT AS A CONTROLLED LEAK IF LOW PRESSURE REGULATOR DIAPHRAGM SHOULD FAIL.

NORMALLY THE RESTRICTOR IS A ZERO FLOW ORIFICE.

EXTERNAL LEAKAGE - Zero

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

PROOF PRESSURE - 60 PSIA

BURST PRESSURE - 75 PSIA

MATERIAL - Anodized Aluminum or Stainless Steel.

Item J - Calibrated Orifice

INLET TEMPERATURE - $0^{o}C$ to $60^{o}C$

FLOW - 15 std. $in.^{3}$/min. nitrogen gas, Flow decreases as inlet pressure decreases.

INLET PRESSURE - 3000 PSIA - as reservoir exhausts pressure decreases to 300 PSIA

OUTLET PRESSURE - 12 $\pm$ 1/2 PSIA to 15 $\pm$ 1/2 PSIA

LIFE - 10,000 hours

PROOF PRESSURE - 4500 PSIA

BURST PRESSURE - 6000 PSIA

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

EXTERNAL LEAKAGE - Zero

MATERIAL - Anodized Aluminum or Stainless Steel

Item L - High Pressure Filter

TEMPERATURE - $0^{o}C$ to $60^{o}C$

INPUT PRESSURE - 3000 PSI maximum

CLEAN PRESSURE DROP - 1.0 PSI maximum at flow of 15 std. $in.^{3}$/min. nitrogen gas

REMOVAL RATING - 98% 1.5 microns
100% 15.0 microns

EXTERNAL LEAKAGE - Zero

PROOF PRESSURE - 4500 PSIA

BURST PRESSURE - 6000 PSIA

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

MATERIAL - Anodized aluminum or stainless steel

SURFACE AREA OF FILTER OPEN TO GAS FLOW TO BE 3.0 $IN.^{2}$ MINIMUM.

FILTER MAY BE INCORPORATED IN A COMMON MANIFOLD CONTAINING ITEMS V, N, AND T.

Item M - Pressure Switch

SWITCH ACTUATES AT 3000 $^{+200}_{-000}$ PSIA

SWITCH BREAKS AT LESS THAN 3000 PSIA

TEMPERATURE - $0^{\circ}C$ to $60^{\circ}C$

NITROGEN GAS

AUTOMATIC RESET

28 VDC

HERMETIC ELECTRICAL CONNECTOR - MS-3112, MS-3113

LIFE - 10,000 hours

CYCLES OF OPERATION - 1000

PROOF PRESSURE - 4500 PSIA

BURST PRESSURE - 6000 PSIA

EXTERNAL LEAKAGE - Zero

INLET PORT - AND-10050-4

SWITCH MAY BE BELLOWS, BOURDON TUBE OR METAL DIAPHRAGM CONSTRUCTION.

SWITCH CONTACTS TO BE HERMETICALLY SEALED

ELECTRICAL LOAD ON SWITCH CONTACTS TO BE INDUCTIVE (A RELAY)

MAXIMUM CURRENT WILL BE ONE AMPERE

MATERIAL - Anodized aluminum or stainless steel

Item N - Reservoir Fill Valve

TWO VALVES REQUIRED LOCATED AT RIGHT ANGLES

NITROGEN GAS

TEMPERATURE - $0^{\circ}C$ to $60^{\circ}C$

LIFE - 10,000 hours

INLET PRESSURE - 3000 PSIA

MINIMUM SEATING PRESSURE - 35 PSIA for zero leakage

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4 (IF REQUIRED)

EXTERNAL LEAKAGE - Zero 35 to 3000 PSIA

PROOF PRESSURE - 4500 PSIA

BURST PRESSURE - 6000 PSIA

FAIL CLOSE

MATERIAL - Anodized aluminum or stainless steel.

Item P - Reservoir

PRESSURE - 3000 $^{+200}_{-000}$ PSIA

TEMPERATURE - $0^{o}C$ to $60^{o}C$

NITROGEN GAS

VOLUME - 1/2 $ft^3$ (APPROX.)

MATERIAL - Electroless nickel-plated steel

INLET PORT - AND-10050-4 minimum

LIFE - 10,000 hours

EXTERNAL LEAKAGE - Zero

PROOF PRESSURE - 4500 PSIA

BURST PRESSURE - 6000 PSIA

Item R - Relief Valve

INLET PRESSURE - 12 to 15 PSIA

RELIEF PRESSURE SETTING - 18 $\pm$ 1 PSIA

INLET TEMPERATURE - $0^{o}C$ to $60^{o}C$

NITROGEN GAS FLOW - 15 std. $in.^3$/min. minimum when open at 19 PSIA

LEAKAGE WHEN CLOSED - Zero

LIFE - 10,000 hours

INLET PORT - AND-10050-4

VALVE TO OPERATE NORMALLY AFTER BEING SUBJECTED TO 30 PSIA

MATERIAL - Anodized aluminum or stainless steel.

Item S - Absolute Pressure Regulator

INLET TEMPERATURE - $0^{o}C$ to $60^{o}C$

INLET PRESSURE - 3000 PSIA - as reservoir exhausts pressure decreases to 300 PSIA

OUTLET PRESSURE - 13.5 $\pm$ 0.5 PSIA

FLOW - 15 std. $in.^3$/min. nitrogen gas

EXTERNAL LEAKAGE - Zero

INTERNAL LEAKAGE - Zero

LIFE - 10,000 hours

INLET PORT - AND-10050-4

OUTLET PORT - AND-10050-4

PROOF PRESSURE AT INLET - 4500 PSIA

PROOF PRESSURE AT OUTLET - 30 PSIA

BURST PRESSURE AT INLET - 6000 PSIA

BURST PRESSURE AT OUTLET - 45 PSIA

VALVE FAILS OPEN

MATERIAL - Anodized aluminum or stainless steel

Item T - Relief Valve (Reservoir Protection)

INLET PRESSURE - 0 to 3000 PSIA

OPENING PRESSURE - 3300 PSIA

RESEAT PRESSURE - 3150 PSIA

INLET TEMPERATURE - $0^{o}C$ to $60^{o}C$

LEAKAGE WHEN SEATED - Zero

LIFE - 10,000 hours

INLET PORT - AND-10050-4, or valve may be part of common manifold for items V, N, and T.

MATERIAL - Anodized aluminum or stainless steel

Item V - Burst Disk (Reservoir Protection)

INLET PRESSURE - 0 to 3000 PSIA

INLET TEMPERATURE - $0^{o}C$ to $60^{o}C$

LEAKAGE - Zero

LIFE - 10,000 hours

BURST PRESSURE - 3900 $\pm$ 5% PSIA

MATERIAL - Anodized aluminum or stainless steel

INLET PORT - AND-10050-4, may be part of common manifold for items V, N, and T.

## REVISIONS

| REV. | DESCRIPTION | CHANGE BY | DATE | APPROVAL BY | DATE |
|---|---|---|---|---|---|
| | | | | | |

DOCUMENT NO. TR-670

PROJECT NO. D-670

APPENDIX C

| APPROVAL | DATE | PROJECT NO. D-670 | AERODYNE CONTROLS CORPORATION FARMINGDALE, NEW YORK |
|---|---|---|---|
| PROGRAM APPROVAL | | R.F.Q. NO. 65-1-19-1 | |
| RELIABILITY APPROVAL | | CONTRACT NO. 65-336 | TITLE ACCEPTANCE TEST REPORT MAKE-UP GAS SUPPLY SYSTEM P/N 670-1-000 |
| Q.C. APPROVAL | | CUSTOMER MECHANICAL TECHNOLOGY INC LATHAM, NEW YORK | |
| PROJECT ENG. | | | SIZE A / CODE IDENT. NO. / DOCUMENT NO. TR-670 / REV |
| CHECKED | | | |
| PREPARATION | | | SCALE / WEIGHT / SHEET 1 OF 10 |

## INTRODUCTORY COMMENTS

The tests reported in this document were conducted in Aerodyne Controls environmentally controlled "white room". The test gas used was specially conditioned, dry, oil-free air prefiltered to a ten (10) micron absolute level.

Typical test conditions were as follows:

1. Instrumentation accuracy

   Unless otherwise specified
   $\pm$ 1/2% for all pressures and
   $\pm$ 2% for all flows

2. Test specimen ambient: white room area

   70$\pm$2°F, 33 to 34 inches mercury pressure
   and 50% or less relative humidity

3. Instrumentation ambient

   70$\pm$10°F, 28 to 32 inches mercury pressure
   and 80% or less relative humidity

## 1.0 SUMMARY

This report presents the results of final acceptance testing on a Make-up Gas Supply System, Aerodyne Controls Corporation P/N 670-1-000, designed to meet or better the requirements of Mechanical Technology Incorporated Component Definition Sheets (Ref: MTI Quotation Request No. 65-1-19-1, Item 2).

The Make-up Gas Supply System contains the following components (Ref: Aerodyne Controls Dwg. 670-1-000B).

a) Storage Vessel

b) Manifold Assembly

1. Limiting orifice, charging flow
2. Two (2) identical charging valves 90° out of phase with respect to location
3. Relief Valve
4. Rupture Disc
5. Pressure Switch
6. Filter Assembly
7. Flow limiting orifice, storage vessel flow

c) Absolute Pressure Regulator

d) Absolute Pressure Relief Valve

The tests performed and reported in this document are the following:

1. Storage Vessel Proof Pressure
2. Actuation pressure and de-actuation pressure of the pressure switch (Item B-5 above)
3. Cracking and reseat pressure of the manifold relief valve (Item B-3 above)
4. Cracking pressure of the charging valves (Item B-2 above)
5. Flow data on the storage vessel flow limiting orifice (Item B-7 above)
6. Cracking and reseat pressure of the absolute pressure relief valve (Item D above)
7. Regulation characteristics of the absolute pressure regulator (Item C above)

8. Total system external leakage
9. Total system cleanliness

The results of all the above tests were satisfactory.

2.0 TEST RESULTS

2.1 Storage Vessel Proof Pressure

The storage vessel was pressurized 4500 PSIA hydraulically and maintained at this pressure for a duration of five (5) minutes. Mechanical gages mounted on the external surface of the storage vessel for strain detection indicated no yielding or damage. The results of the test were satisfactory.

2.2 Pressure Switch Actuation

The pressure switch, Aero Mechanisms Inc., Model 6455, S/N 118, was pressurized pneumatically to obtain electrical indication of switch closure and hysteresis. The switch closed at 3010 PSIA (±10 PSI gage accuracy) and opened on decreasing pressure at 2990 PSIA. The test results were satisfactory.

2.3 Cracking and Reseat Pressure of the Manifold Relief Valve

The redesigned manifold relief valve was pressurized pneumatically. The cracking pressure of the relief valve was 3300 PSIG, indicated by collection of bubbles emanating from the relief valve (which was submerged in isopropyl alcohol) at a rate in excess of 30 SCCM. With 4500 PSIG air applied to the manifold inlet, the storage vessel pressure was allowed to build up to 3300 PSIG at which point the relief valve cracked. Storage vessel pressure rose to 3400 PSI (full flow condition for the relief valve) and then began to decay. The relief valve reseated at a manifold (storage vessel) pressure of 600 PSIG. External leakage at the 600 PSIG pressure was measured by bubble collection with the relief valve submerged. No bubbles were collected in a three (3) minute period. Results of the test were satisfactory.

2.4 Cracking Pressure of Charging Valves

A low pressure air supply was connected to the manifold inlet. The first charging valve was installed; the second charging valve was not. A pressure of 33 PSIG was supplied to the manifold inlet port. Collecting bubbles, with the manifold submerged in isopropyl alcohol, a flow rate of 1.0 SCCM was indicated. The pressure applied to the manifold inlet port was decreased to 30 PSIG. No bubbles were collected during the three (3) minute collection period.

The second charging valve was installed in the manifold and the manifold inlet pressurized. Cracking pressure was indicated by collecting bubbles emanating from the pressure switch port (the pressure switch was not yet installed on the manifold) with the manifold submerged in isopropyl alcohol. With an inlet pressure of 43 PSIG applied to the manifold inlet port, a flow of 1.0 SCCM was collected. Decreasing the manifold inlet pressure to 35 PSIG, no bubbles were collected for the three (3) minute collection period.

The air supply was then disconnected from the manifold inlet port and attached to the pressure switch port. A pressure of 35 PSIG was applied to the pressure switch port. With the manifold submerged, no bubbles were collected over a ten minute period. This indicated that the charging valves setting was sufficient to hold a storage vessel pressure of 35 PSIG.

The charging valve test results were satisfactory.

2.5 Limiting Orifice Flow Data

Flow data on the flow limiting orifice was taken prior to assembly of the limiting orifice in the manifold. With the limiting orifice (actually a capillary tube assembly) mounted in a special test fixture, an inlet pressure of 3200 PSIG (air) was applied. The limiting orifice flow rate, measured on a Fisher & Porter flow-meter (2% accuracy), was 265 SCCM which is equivalent to 16.27 SCIM. The flow rate thru the assembly was also measured at inlet pressures of 3000, 2500, 2000, 1500, 500 and 300 PSIG. The results of these measurements are presented graphically in Figure I (see attachment).

2.6 Absolute Pressure Relief Valve

The absolute pressure relief valve, Aerodyne P/N 672-2-000, S/N 001, was tested separate from the system. A low pressure air supply was attached to the inlet of the relief valve, and a pressure of 30 PSIA applied for a period of five (5) minutes during which time the valve was relieving.

The unit was then submerged in isopropyl alcohol for determination of the cracking pressure by bubble collection. Starting at 15 PSIA the inlet pressure to the unit was slowly increased. A flow rate of 1.0 SCCM was found at an applied inlet pressure of 18.3 PSIA (measured on a manometer, $\pm .02 \frac{PSIA}{accuracy}$). The inlet pressure was increased to 19 PSIA and the flow measured at this condition was in excess of 30 SCIM. The inlet pressure was slowly decreased. The relief valve reseated at 17.4 PSIA, as indicated by no bubbles emanating from the relief valve vent port for a period of three (3) minutes. There was no external leakage (which would have been indicated by bubbles) at the 17.4 PSIA inlet pressure when held for a period in excess of 10 minutes. The test results on P/N 672-1-000 were satisfactory.

2.7 Absolute Pressure Regulator (P/N 674-1-000)

The absolute pressure regulator, Aerodyne P/N 674-1-000 was proof pressure tested with an applied inlet pressure of 4500 PSIA held for a period of five (5) minutes. No external leakage was visible with the unit submerged in isopropyl alcohol. The regulated outlet pressure during the test was 13.1 PSIA.

The inlet pressure to the unit was decreased to 3000 PSIA and held for a period in excess of five (5) minutes. Simultaneously, the outlet port was pressurized to 18 PSIA and held for the five minute period. During the test, the unit was submerged in isopropyl alcohol. No bubbles, (indicating either internal or external leakage) were found.

The outlet pressure air supply was removed and the unit removed from the isopropyl alcohol. The units regulation characteristics were tested with dry air at inlet pressures of 3000, 2500, 2000, 1500, 1000, 500 and 300 PSIA and flow rates of 15 SCIM and zero flow. The outlet pressure varied between 13.30 PSIA and 13.68 PSIA (measured on a manometer, 0.02 PSIA accuracy), well within the allowable tolerance of 13.5±0.5 PSIA.

The test results were satisfactory.

2.8 Total System External Leakage

The complete 670-1-000A system was checked for external leakage. The storage vessel was charged to 3000 PSIG and the inlet supply disconnected. The system was submerged in a tank of isopropyl alcohol for five (5) minutes. The absolute pressure regulator, P/N 674-1-000, was in a zero flow condition, since the outlet port was vented to ambient (14.7 PSIA). No bubbles, indicative of external leakage, were observed emanating from the system surfaces and seals. The system was removed from the isopropyl alcohol and the pressure switch external leakage seal broken by partial backing off of the pressure switch from the manifold face, thus venting the storage vessel charge. The pressure switch was then returned to its sealing position and lockwired.

## 2.9 Total System Cleanliness

System cleanliness was determined by Gas Scrubber Test on 1) P/N 674-1-000, absolute pressure regulator, and P/N 672-2-000 absolute pressure relief valve, connected together (as in the system) and 2) on the complete manifold assembly, with storage vessel, but with the flow limiting orifice not installed.

These tests were conducted prior to final assembly since it was necessary for 1) the outlet pressure of the absolute pressure regulator P/N 674-1-000 to be raised to 16.5 PSIA to obtain the required gas scrubber test flow and 2) to conduct the gas scrubber test on the manifold assembly without the flow limiting orifice installed in order to meet the required gas scrubber test flow requirement. Results of both tests were satisfactory.

The absolute pressure regulator P/N 674-1-000 was set at its proper outlet pressure and a regulation test performed on the unit (see Para. 2.7). The storage vessel was removed from the manifold and the flow limiting orifice installed. The storage vessel was re-assembled on the manifold, P/N 674-1-000, absolute pressure regulator, and P/N 672-2-000, absolute pressure relief valve, were assembled to the manifold and the external leakage test (see Para. 2.8) conducted.

# ATTACHMENTS

1. Test Data Sheet dated 12-1-65, P/N 670-1-000A
2. Test Data Sheet dated 12-1-65, P/N 674-1-000
3. Figure I - Manifold Capillary Assy., Gas Bearing Makeup, Gas Supply System, P/N 670-1-000

| CODE IDENT. NO. | DWG. SIZE | DOCUMENT NO. | REV. |
|---|---|---|---|
| | A | TR-670 | |
| | | SHEET 10 OF 10 | |

**AERODYNE CONTROLS CORPORATION**
90 Gazza Boulevard
Farmingdale, N.Y.

Pg._____ of _____

Units Tested System Components
Date 12-1-65
P/N 670-1-000A
S/N — Tests Listed Below
Tester E. Shea
Insp. R. Tripodi

Test Fluid Dry air Environment Clean Room

| Component | Part No. Serial No. | Inlet Pressure | Flow SCIM | Response | Time | Remarks |
|---|---|---|---|---|---|---|
| | | | | | | |
| Relief Valve | 672-2-000 001 | 18.3 PSIA | Cracks | | | |
| | | 19 PSIA | >>30 | | | |
| | | 17.4 PSIA | Reseat | | | |
| | | | | | | |
| Storage vessel | | 4500 PSIG | | | 5 min. | No Damage or yielding |
| | | | | | | |
| Manifold Relief Valve | | 3300 PSIG | Cracks | | | |
| | | 3400 PSIG | Full flow | | | |
| | | 600 PSIG | Reseat | | | |
| Manifold Charging Valve | 1st | 33 PSIG | Crack | | | |
| | 2nd | 43 PSIG | Crack | | | |
| | | | | | | |
| Manifold Press. Switch | | 3010 PSIA | | actuates | | |
| | | 2990 PSIA | | Breaks | | |
| System | | 3000 PSIG | | | | No External Leakage |
| | | | | | | |
| | | | | | | |
| | | | | | | |

**NOTES:**

**AERODYNE CONTROLS CORPORATION**
**90 Gazza Boulevard**
**Farmingdale, N.Y.**

Pg____of____

**Date** 12-1-65

**Unit Tested** Regulator

**Tests** Proofpressure, Leakage, and Regulation

**Tester** E. Shea

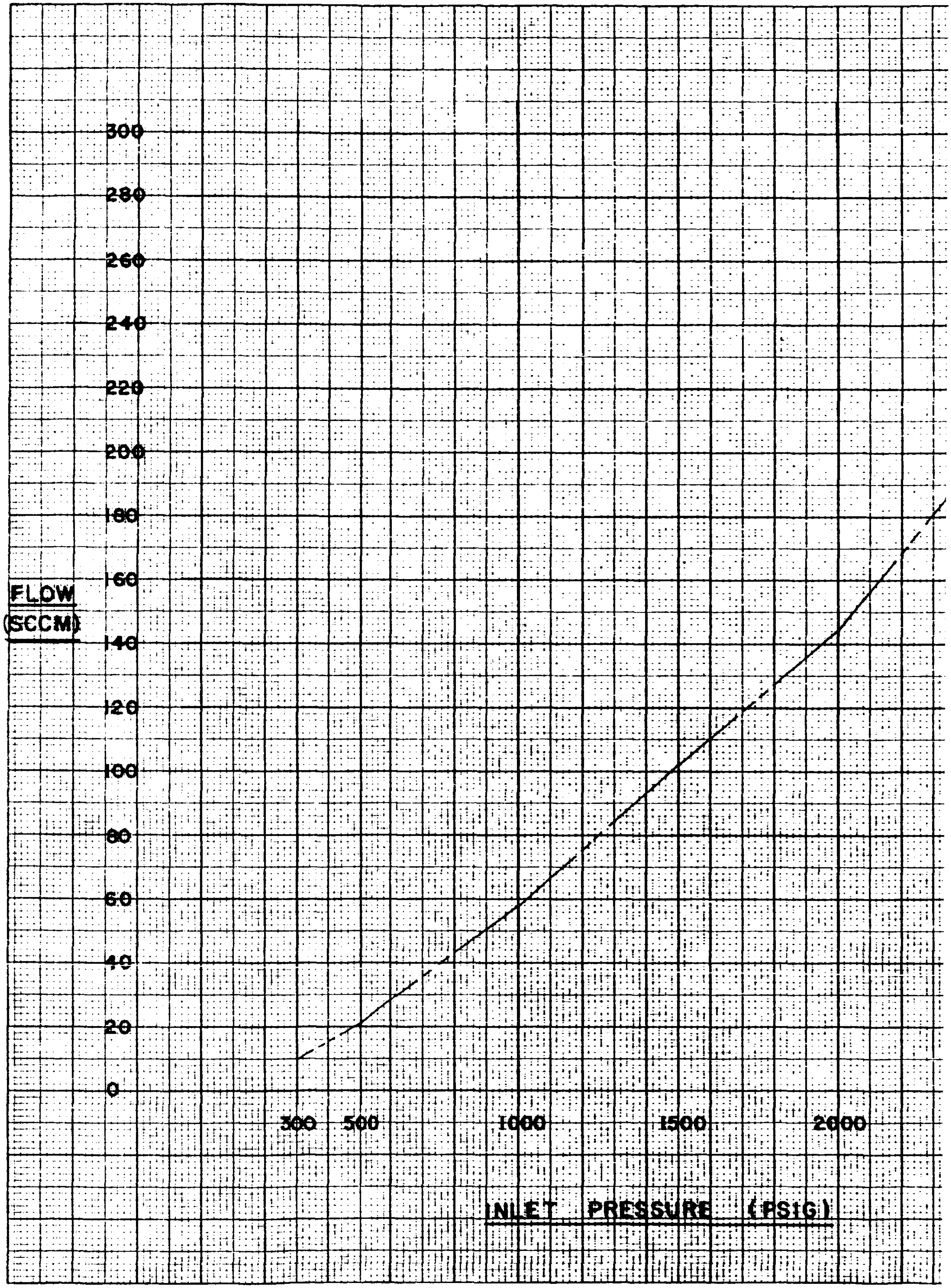
**P/N** 674-1-000

**Test Fluid** Dry air **Environment** Clean Room **Insp.** R. Tripodi

**S/N** 001

| Test | Inlet Pressure | Outlet Pressure | Flow Reading | Actual Flow | Time of Test | Remarks |
|---|---|---|---|---|---|---|
| | PSIA | PSIA | | SCIM | | |
| | | | | | | |
| Proof Pressure | 4500 | 13.1 | 5.16 | 15 | 5 mins. | No damage |
| External Leakage | 3000 | 18 | 0 | 0 | 5 mins. | No Leakage |
| Internal Leakage | 0 | 18 | 0 | 0 | 5 mins. | No Leakage |
| Regulation | 3000 | 13.30 | 5.16 | 15 | | |
| | 3000 | 13.33 | 0 | 0 | | |
| | 2500 | 13.30 | 5.16 | 15 | | |
| | 2500 | 13.33 | 0 | 0 | | |
| | 2000 | 13.40 | 5.16 | 15 | | |
| | 2000 | 13.40 | 0 | 0 | | |
| | 1500 | 13.45 | 5.16 | 15 | | |
| | 1500 | 13.48 | 0 | 0 | | |
| | 1000 | 13.53 | 5.16 | 15 | | |
| | 1000 | 13.55 | 0 | 0 | | |
| | 500 | 13.60 | 5.16 | 15 | | |
| | 500 | 13.63 | 0 | 0 | | |
| | 300 | 13.65 | 5.16 | 15 | | |
| | 300 | 13.68 | 0 | 0 | | |

**NOTES:** Internal Leakage measured at inlet port


FLOW
(SCCM)
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
300
500
1000
1500
2000
INLET PRESSURE (PSIG)

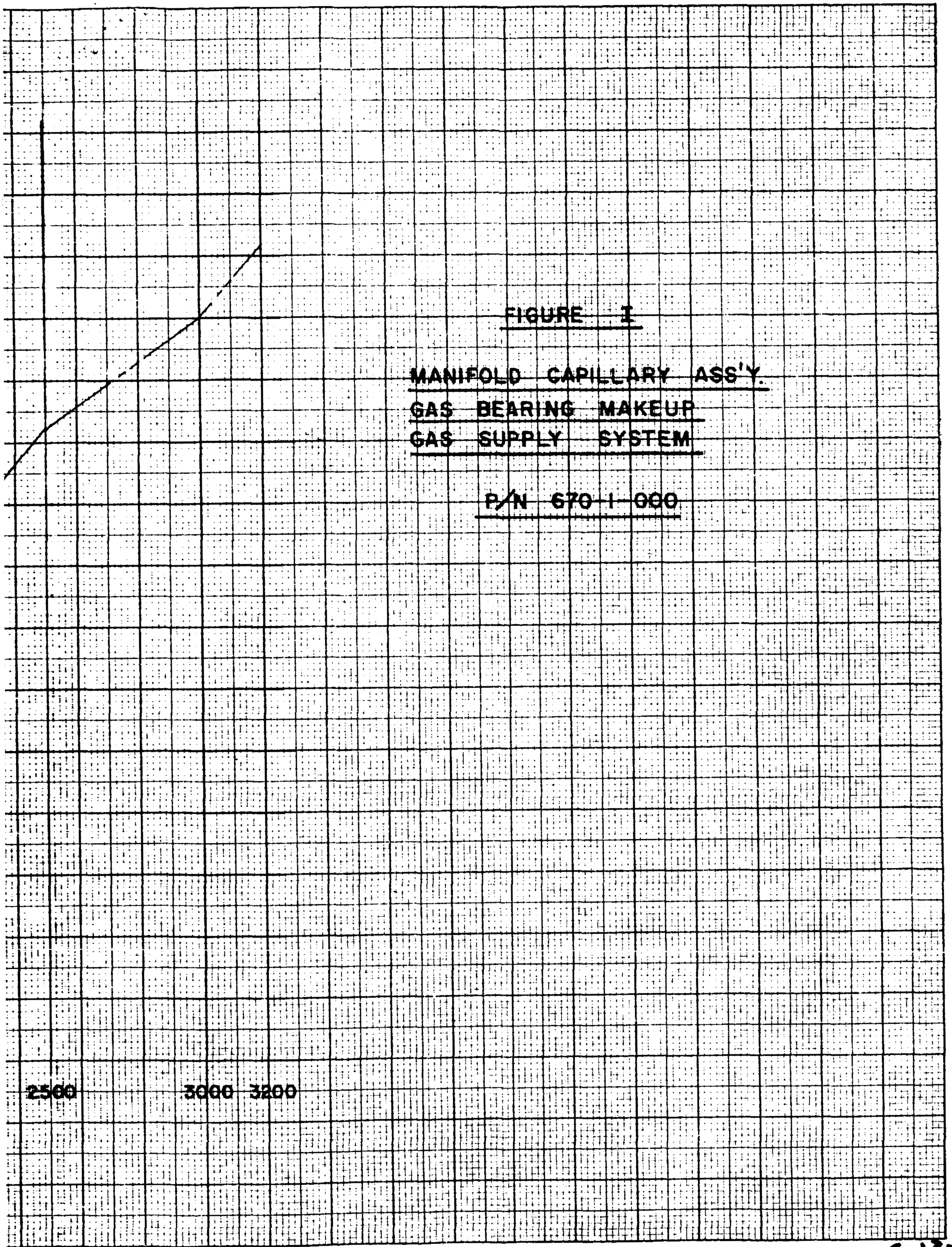



FIGURE I

MANIFOLD CAPILLARY ASS'Y.
GAS BEARING MAKEUP
GAS SUPPLY SYSTEM

P/N 670-1-000

# APPENDIX D

# PARTS LISTS

MECHANICAL TECHNOLOGY INCORPORATED

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sheet 1 of 4

Title PARTS LIST FOR PANEL ASSEMBLY GAS SUPPLY SYSTEM

PL141F28
sheet 1 of 4

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | X | 1 | ASSEMBLY | 141F28G1 |
| | | | | | 1 | 2 | ASSEMBLY | PL141F20G1 |
| | | | | | | | MOUNTING | |
| | | | | | | | (RESONANT | |
| | | | | | | | PISTON COMPR.) | |
| | | | | | 1 | 3 | BRACKET | 141C29G1 |
| | | | | | | | (TUBE) | |
| | | | | | 2 | 4 | BRACKET | 141B30P1 |
| | | | | | | | (FILTER) | |
| | | | | | 1 | 5 | BRACKET | 141B31P1 |
| | | | | | | | (REGULATOR) | |
| | | | | | 1 | 6 | BRACKET | 141C32G1 |
| | | | | | | | (CONNECTOR) | |
| | | | | | 1 | 7 | BRACKET | 141C33P1 |
| | | | | | | | (GAS SUPPLY | |
| | | | | | | | CONTAINER) | |
| | | | | | 1 | 8 | MOUNTING | 141E34G1 |
| | | | | | | | (GAS SUPPLY | |
| | | | | | | | CONTAINER) | |
| | | | | | 1 | 9 | SUPPORT | 141C35G1 |
| | | | | | | | (GAS SUPPLY | |
| | | | | | | | CONTAINER) | |
| | | | | | 1 | 10 | METAL HOSE | 141B36G1 |
| | | | | | | | ASSEMBLY | |
| | | | | | 1 | 11 | RETAINING | 141B37P1 |
| | | | | | | | CLIP | |
| | | | | | 2 | 12 | STRAP | 141D38P1 |
| | | | | | 1 | 13 | CABLE ASSY. | 141C39G1 |
| | | | | | | | ELECTRICAL | |

1 PTG DWG SIZE WAS B FFB 6/22

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Checked H. Jones 3/29/65
Drawn H. Bartmeker 15 MAR '65

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Title PARTS LIST FOR
PANEL ASSEMBLY
GAS SUPPLY SYSTEM

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 1 | 14 | DRIVE CIRC- | 141D40G1 |
| | | | | | | | UIT ASSEMBLY | |
| | | | | | | | | |
| | | | | | 1 | 15 | REGULATOR | DWG. NO. 653-1-000 |
| | | | | | | | LOW PRESSURE | AERODYNE CONTROLS CORP. |
| | | | | | 1 | 16 | FILTER | CAT.# ACI 3105-42A |
| | | | | | | | LOW PRESSURE | AIRCRAFT POROUS MEDIA INC. |
| | | | | | 2 | 17 | CLAMP | CAT.# TA4C74WD9T (ANODIZE) |
| | | | | | | | TEFLON | TA MFG. CORP. OR EQUIV. |
| | | | | | | | CUSHION | |
| | | | | | 4 | 18 | CLAMP | CAT.# TA4C74WD4T (ANODIZE) |
| | | | | | | | TEFLON | TA MFG. CORP OR EQUIV. |
| | | | | | | | CUSHION | |
| | | | | | 2 | 19 | FLARED TUBE | CAT.# 6F5BX-SS |
| | | | | | | | FITTING | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 20 | METAL HOSE | CAT. # 351C-4-B2B2-06-0184 |
| | | | | | | | ASSEMBLY | FLEXONICS OR EQUIV. |
| | | | | | 2 | 21 | METAL HOSE | CAT.# 351C-4-D2B2-06-0394 |
| | | | | | | | ASSEMBLY | FLEXONICS OR EQUIV. |
| | | | | | 2 | 22 | FLARED TUBE | CAT.# 4F5BX-SS |
| | | | | | | | FITTING | PARKER OR EQUIV. |
| | | | | | 2 | 23 | STRAIGHT | CAT.# 4C5BX-SS |
| | | | | | | | THREAD ELBOW | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 24 | STRAIGHT | CAT.# 6FF5BX-SS |
| | | | | | | | THREAD | PARKER OR EQUIV. |
| | | | | | | | LONG CONN. | |
| | | | | | 1 | 25 | REDUCER | AN919-6D |
| | | | | | | | EXT. THREAD | |

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Title PARTS LIST FOR
PANEL ASSEMBLY
GAS SUPPLY SYSTEM

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 1 | 26 | SLEEVE, | AN 819-6D |
| | | | | | | | COUPLING | |
| | | | | | 1 | 27 | NUT, | AN 818-6D |
| | | | | | | | COUPLING | |
| | | | | | 1 | 28 | ADAPTER | CAT.# 4-6 F505G5-SS |
| | | | | | | | | PARKER FITTINGS INC. |
| | | | | | 1 | 29 | STRAIGHT | CAT.# 6C5BX-SS |
| | | | | | | | THREAD ELBOW | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 30 | TEE | CAT.# 6WJJBTX-SS |
| | | | | | | | | PARKER FITTINGS OR EQUIV. |
| | | | | | 2 | 31 | BULKHEAD | 6WEBTX-SS |
| | | | | | | | UNION ELBOW | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 32 | BULKHEAD | CAT.# 6WBTX -SS |
| | | | | | | | UNION | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 33 | BULKHEAD | CAT.# 4 WEBTX-SS |
| | | | | | | | UNION ELBOW | PARKER FITTINGS OR EQUIV. |
| | | | | | 1 | 34 | TUBING | 1/4 O.D. X .020 WALL X 12" LG |
| | | | | | | | | STN. STL. AISI 304 (AMS 5560D) |
| | | | | | 1 | 35 | TUBING | 3/8 O.D. X .020 WALL X 12" LG |
| | | | | | | | | STN. STL. AISI 304 (AMS 5560D) |
| | | | | | 1 | 36 | TUBING | 1/4 O.D. X .020 WALL X 15" LG |
| | | | | | | | | STN. STL. AISI 304 (AMS 5560D) |
| | | | | | 1 | 37 | TUBING | 1/4 O.D. X .020 WALL X 40" LG |
| | | | | | | | | STN. STL. AISI 304 (AMS 5560D) |
| | | | | | 1 | 38 | TUBING | 1/4 O.D. X .020 WALL X 16" |
| | | | | | | | | STN. STL. AISI 304 (AMS 5560D) |
| | | | | | 1 | 39 | STRAIGHT THRU | CAT.# 4R5BX-SS |
| | | | | | | | RUN TEE | PARKER FITTINGS OR EQUIV. |

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Title PARTS LIST FOR
PANEL ASSEMBLY
GAS SUPPLY SYSTEM

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 2 | 44 | MOUNT | 141B46P1 |
| | | | | | | | (FILTER) | |
| | | | | | 4 | 45 | HEX. SOCKET | 1/4-20 X 3 LG. |
| | | | | | | | HD. CAP SCR. | STN. STL. |
| | | | | | 17 | 46 | HEX. SOCKET | 1/4-20 X 1" LG |
| | | | | | | | HD. CAP SCR. | STN. STL. |
| | | | | | 4 | 47 | SHOULDER | 3/8 X 1 1/2 LONG (REGULAR) |
| | | | | | | | SCREW | ALLOY STL. (CAD PLATE) (MS 16638-21) |
| | | | | | 4 | 48 | HEX. SOCKET | #10-32 X 3/4 LONG |
| | | | | | | | HD. CAP SCR | STN. STL. |
| | | | | | 2 | 49 | HEX. SOCKET | 1/4-20 X 1 3/4 LG |
| | | | | | | | HD. CAP. SCR. | STN STL. |
| | | | | | 4 | 50 | NUT | CAT. # 79NE-040 |
| | | | | | | | | ESNA OR EQUIV. |
| | | | | | 4 | 51 | NUT | CAT. # 79 NM-02 |
| | | | | | | | | ESNA OR EQUIV |
| | | | | | 4 | 52 | NUT | CAT. # 42NE-058 |
| | | | | | | | | ESNA OR EQUIV. |
| | | | | | 25 | 53 | WASHER | AN 960-C416L |
| | | | | | 8 | 54 | WASHER | AN 960-616L |
| | | | | | 8 | 55 | WASHER | AN 960-C10L |
| | | | | | 2 | 56 | HEX. SOCKET | 1/4-20 X 7/8 LG |
| | | | | | | | HD. CAP. SCR | STN. STL. |

1 PT. 51 WAS CAT# 79NE-02 F.E.B.

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sheet 1 of 2

Title
LAYOUT - MOUNTING
RESONANT PISTON COMPRESSOR

| sheet 6 | 5 | 4 | 3 | 2 | 1 (of REQ'D) | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 1 | 1 | COMPRESSOR | 141F02 |
| | | | | | 6 | 2 | SPRING-EXT. | ASSOCIATED SPRING CORP. No |
| | | | | | | | | E750-075-250. STAINLESS |
| | | | | | | | | STEEL [FED. SPEC. QQ-W-423, COMP |
| | | | | | | | | FS302 OR FS304] AMS 5688. |
| | | | | | | | | SPRING TEMPER. |
| | | | | | | | | |
| | | | | | 2 | 3 | SHOCK MOUNT | ROBINSON TECH. PRODUCTS INC. |
| | | | | | | | | TETERBORO, N.J. MODEL No |
| | | | | | | | | W681-5 OR EQUIV MODEL No |
| | | | | | | | | OF 7002 SERIES (MODIFIED BY |
| | | | | | | | | MTI) |
| | | | | | 12 | 4 | HOLDER-SPRING | AMS 5610E HEX. STOCK Rc 32-36 |
| | | | | | | | | 141B26 |
| | | | | | 1 | 5 | FRAME | WELDED FABRICATION |
| | | | | | | | | AL. ALLOY 6061-T6 |
| | | | | | | | | ANODIZE AMS 2470D |
| | | | | | | | | 141E41 |
| | | | | | 2 | 6 | SPRING-BAR | AMS 5610E Rc 36-40 |
| | | | | | | | | 141B27 |
| | | | | | 8 | 7 | LUG-MOUNTING | AL. ALLOY 6061-T6 |
| | | | | | | | | ANODIZE AMS 2470D |
| | | | | | | | | 141B42 |
| | | | | | 4 | 8 | LUG-MOUNTING | " " |
| | | | | | | | | 141B43 |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | 1 | 10 | SHOCK MOUNT | AS FOR PART 3 |
| | | | | | | | | |

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Checked H. JONES 3/29/65
Drawn H. JONES 1/28/65

Rev. No.

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sheet 1 of 2

MECHANICAL TECHNOLOGY INCORPORATED

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Title
LAYOUT- MOUNTING
RESONANT PISTON COMPRESSOR

sheet of REQ'D

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 37 | 14 | WASHER | AN960-PD416 - AL. ALLOY |
| | | | | | | | | 1/4 DIA SCREW BY 1/16 THICK |
| | | | | | 6 | 15 | SCREW | 1/4 - 20 UNC-3A CAP-CYL. SOCKET |
| | | | | | | | 1" LG. | HD., HEX., STEEL (CAD PLATE) |
| | | | | | | | NYLOK | |
| | | | | | 16 | 16 | SCREW | " " |
| | | | | | | | 1 LG. | |
| | | | | | | | NYLOK | |
| | | | | | 6 | 17 | SCREW | " " |
| | | | | | | | 1 1/2 LG | |
| | | | | | 6 | 18 | SCREW | " " |
| | | | | | | | 2 LG | |
| | | | | | 24 | 20 | LOCK NUT | 1/4 -20 UNC -3B |
| | | | | | | | | ESNA Nº 68 NE-040 |
| | | | | | 3 | 21 | SCREW | 1/4-28 UNF-3A CAP-CYL., SOCKET HD., HEX. |
| | | | | | | | 3/4 LG. NYLOK | STEEL (CAD PLATE) |

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Rev. No.

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Title PARTS LIST FOR
ASSEMBLY
RESONANT PISTON COMPRESSOR

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | X | 1 | ASSEMBLY | 141F02G1 |
| | | | | | 1 | 2 | CONTAINER | 141E21G1 |
| | | | | | | | ASSEMBLY | |
| | | | | | 1 | 3 | SPACER | 141B05P1 |
| | | | | | 1 | 4 | SPRING | 141D01G1 |
| | | | | | | | ASSEMBLY | |
| | | | | | 1 | 5 | END CAP | 141B06P1 |
| | | | | | 1 | 6 | ASSEMBLY | 141B22G1 |
| | | | | | | | CYLINDER & | |
| | | | | | | | VALVE BLOCK | |
| | | | | | 1 | 7 | PISTON | 141B23G1 |
| | | | | | | | ASSEMBLY | |
| | | | | | 2 | 8 | MOUNT | 141B19P1 |
| | | | | | | | (CYLINDER) | |
| | | | | | 2 | 9 | MOUNTING | 141C16G1 |
| | | | | | | | ASSEMBLY | |
| | | | | | | | (SOLENOID) | |
| | | | | | 2 | 10 | MOUNTING | 141C16G2 |
| | | | | | | | ASSEMBLY | |
| | | | | | | | (SOLENOID) | |
| | | | | | 1 | 11 | SPACER | 141B50P1 |
| | | | | | | | VALVE | |
| | | | | | 1 | 12 | STOP VALVE | 141B14P1 |
| | | | | | 1 | 13 | VALVE | 141B49P1 |
| | | | | | 1 | 14 | COVER | 141C11P1 |
| | | | | | 1 | 15 | PLUNGER | 141B24G1 |
| | | | | | | | (SOLENOID) | |
| | | | | | 1 | 16 | SOLENOID | 141D25G1 |
| | | | | | | | ASSEMBLY | |

| Rev. | Change |
|---|---|
| 1 | PT. 11 DWG NO WAS 141B13P1<br>PT. 13 DWG NO. WAS 141B12P1 6/22 FEB |

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Drawn F. Bartnicki 10 FEB '65

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Title PARTS LIST FOR
ASSEMBLY
RESONANT PISTON COMPRESSOR

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 4 | 17 | SHIM | 141B47P1 |
| | | | | | 1 | 18 | RING | 141B51P1 |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | 1 | 20 | SEAL | GASK-O-SEAL CAT.# 420-015-14 |
| | | | | | | | CONNECTOR | PARKER OR EQUIV. |
| | | | | | 1 | 21 | RECEPTACLE | CAT.# DM5606-194P-013 |
| | | | | | | | | DEUTSCH CO. OR EQUIV. |
| | | | | | 1 | 22 | PLUG | CAT.# DS07-194S |
| | | | | | | | | DEUTSCH CO. OR EQUIV. |
| | | | | | 1 | 23 | TUBING | 3/8 O.D. X .035 WALL X 18" LONG |
| | | | | | | | | AL. ALLOY 6061T6 |
| | | | | | 1 | 24 | TUBING | 3/8 O.D. X .035 WALL 12" LONG |
| | | | | | | | | AL. ALLOY 6061T6 |
| | | | | | 4 | 25 | ADAPTER, | MC237D6 |
| | | | | | | | FLARED TUBE | |
| | | | | | | | & BOSS | |
| | | | | | 4 | 26 | NUT, COUPLING | MC124D6 |
| | | | | | 4 | 27 | SLEEVE, | |
| | | | | | | | FLARED TUBE | MC125D6 |
| | | | | | | | FITTING | |

| | |
|---|---|
| 1 | ADDED PT. 18 FEB 6/22 |

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Title PARTS LIST FOR ASSEMBLY RESONANT PISTON COMPRESSOR

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | | | | |
| | | | | | | | | |
| | | | | | 8 | 33 | HEX. SOCKET | #10-32 X 5/8 LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | | | (DRILLED HD.) | |
| | | | | | 6 | 34 | HEX. SOCKET | #10-32 X 1/2 LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | | | (DRILLED HD) | |
| | | | | | 2 | 35 | HEX. SOCKET | #10-32 X 1/2 LONG (NYLOK) |
| | | | | | | | FLAT HD. | STEEL (CAD PLATE) |
| | | | | | | | SCREW | |
| | | | | | 4 | 36 | HEX. SOCKET | # 5-40 X 5/16 LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | 2 | 37 | HEX-SOCKET | #6-32 X 3/8 LONG |
| | | | | | | | FLAT HD. | |
| | | | | | | | SCREW | |
| | | | | | 4 | 38 | HEX. SOCKET | #10-32 X 5/8 LONG |
| | | | | | | | HD. CAP SCR. | (STEEL CAD PLATE) |
| | | | | | 8 | 39 | HEX. SOCKET | 1/4-20 X 5/8 LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | 2 | 40 | HEX. SOCKET | #8-32 X 3/8 LG. |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | | | (DRILLED HD.) | |
| | | | | | 4 | 41 | HEX. SOCKET | #4-40 X 3/8 LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | | | (DRILLED HD) | |
| | | | | | 4 | 42 | HEX. SOCKET | #10-32 X 2" LONG |
| | | | | | | | HD. CAP SCR. | STEEL (CAD PLATE) |
| | | | | | | | | |

| | | |
|---|---|---|
| 1 | PT 36 WAS #4-40 | FFB 3/4/65 |
| 2 | CHANGED PT-36 TO AGREE WITH REV.-1 | FFB 3/16/66 |

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Title PARTS LIST FOR
ASSEMBLY
RESONANT PISTON COMPRESSOR

| 6 | 5 | 4 | 3 | 2 | 1 | Part No. | Name | Description |
|---|---|---|---|---|---|---|---|---|
| | | | | | 4 | 46 | HEX. NUT | #10-32 STL. (CAD PLATE) |
| | | | | | 8 | 47 | NUT | CAT. # 68NM-02 |
| | | | | | | | | ESNA OR EQUIV. |
| | | | | | 8 | 48 | NUT | CAT. # 68 NE-040 |
| | | | | | | | | ESNA OR EQUIV. |
| | | | | | 16 | 53 | WASHER | AN 960 PD 10L |
| | | | | | 16 | 54 | WASHER | AN 960 PD 416L |
| | | | | | 2 | 55 | HEX. SOCKET | #4-40 X 1/4 LONG |
| | | | | | | | HD. CAP. SCR. | STEEL (CAD PLATE) |

| | | |
|---|---|---|
| 1 | ADDED PT 55 | 9/15/65 FEH |

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Drawn H. Bartnicke 10 FEB. '65

Rev. No. 1

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## NOMENCLATURE

A - area of drive-solenoid plunger pole face

a - single amplitude of piston displacement (one-half the total piston stroke)

b - width of one U-spring element

C - radial clearance between piston and cylinder

E - modulus of elasticity of U-spring material

F - single amplitude of fundamental harmonic of compressor driving force

$\mathfrak{F}$ - total axial force acting on solenoid plunger

f - frequency of piston reciprocation

h - thickness of U-spring element

I - d-c component of solenoid current

i - instantaneous total solenoid current

kp - effective pneumatic spring constant due to piston gas force

L - length of straight section of U-spring element

L - a-c inductance of drive solenoid coil

n - number of U-spring elements

P - total average compression power

p - instantaneous total cylinder pressure

$p_a$ - platform ambient pressure

$\Delta p$ - differential platform pressure; compressor pressure rise

R - radius of U-spring element

R - resistance of drive solenoid coil

t - time

V - instantaneous cylinder volume

w - weight of compressor reciprocating assembly

$x$ - solenoid axial air gap

$\beta$ - flux density in solenoid axial air gap

$\beta$ - taper parameter for self-acting piston bearings

$\delta$ - density of U-spring material

$\eta$ - overall compressor efficiency based on ideal adiabatic compression process

$\sigma$ - U-spring bending stress

$\Phi$ - total flux in solenoid axial air gap

$\omega$ - circular (angular) frequency of piston reciprocation